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SSACV ICEBREAKING LNG TANKER FEASIBILITY STUDY

Prepared for
Transportation Development Centre
Transport Canada

by German & Milne Inc
Naval Architects and Marine Engineers

April 1982

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The contents of this report reflect the view of the Contractor and not necessarily the official views or policies of Transportation Development Centre, Transport Canada.

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17. Abstract

- A comparison is made between a Semi-Submerged Air Cushion Vehicle (SSACV) Icebreaking Liquid Natural Gas (LNG) tanker and a monohull tanker, with the same LNG capacity and power, on the basis of construction cost, annual maintenance cost, manning, and speeds in calm open water and uniform ice.

- For the ideal conditions chosen, and for one level of power, the SSACV tanker is more costly than the monohull vessel in all aspects. It has a slightly lower performance in open water, in which mode the air cushion is not used, but is markedly faster in ice.

- The apparent potential of the concept may be confirmed by model tests, but the practical problems associated with operational use in the Arctic will still be formidable. The economics of specific routes will also have to be considered.

- Model testing is recommended.

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17. Résumé

- Une comparaison est effectuée entre un navire à coques semi-submergées equipé d'un coussin d'air (SSACV) et un méthanier conventionnel ayant la même capacité de méthane liquéfié et la même puissance, en fonction du coût de construction, de l'entretien annuel, du coût de l'équipage, et de la vitesse en eau calme et sur une glace uniforme.

- Pour les conditions idéales choisies, puissance fixe, le méthanier SSACV est plus coûteux que le navire conventionnel.
  Le SSACV a une vitesse légèrement plus basse en eau calme, sans utilisation du coussin d'air, mais il est beaucoup plus rapide dans la glace.

- Le potentiel de ce concept pourrait-être confirmé par des essais sur modèle, mais les problèmes pratiques associés à l'opération dans l'arctique demeurent formidables. Il faudra aussi considérer le rendement économique de routes spécifiques.

- Des essais sur maquettes sont recommandés.

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SUMMARY

A comparison is made between an Semi-Submerged Air Cushion Vehicle (SSACV) icebreaking Liquid Natural Gas (LNG) tanker and a monohull tanker, with the same LNG capacity and power, on the basis of construction cost, annual maintenance cost, manning, and speeds in calm open water and uniform ice.

For the ideal conditions chosen, and for one level of power, the SSACV tanker is more costly than the monohull vessel in all aspects. It has a slightly lower performance in open water, in which mode the air cushion is not used, but is markedly faster in ice.

An LNG tanker is an ice transitting vessel and is not required to break ice other than for its own passage. This SSACV icebreaking tanker requires a full length air cushion to cater for a random requirement of unproven value - rocking the vessel longitudinally to assist icebreaking - and thus appears to be wasting power which could otherwise be used for propulsion. It would seem more beneficial to confine the use of the air cushion to the breaking of ice ahead of the side struts.

The apparent potential of the concept may be confirmed by model tests, but the practical problems associated with operational use in the Arctic will still be formidable. The economics of specific routes will also have to be considered.

Model testing is recommended.

The project team for German & Milne Inc. was composed of:

Mr Nozer Dadachanji
Mr Philip Markham
and others

Dr.TKS Murthy was engaged as an outside SSACV consultant.

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A Letters from British Hovercraft Corporation Limited to Dr. TKS Murthy.

B Telex from British Hovercraft Corporation Limited to German & Milne Inc.

C Minority Report by Dr TKS Murthy

SSACV ICEBREAKING LNG TANKER
FEASIBILITY STUDY

1 INTRODUCTION

An appreciable reduction in resistance may be obtained if a displacement hull can be run below the water surface, because the reduction in wavemaking resistance and wave drag can be greater than the increase in frictional resistance due to the greater wetted surface area. To take advantage of this possibility, various concepts have been evolved from a pure cargo submarine to a semi-submerged vessel with the main hull below the surface and only a thin island emerging. For improved stability, the small waterplane area, twin hull (SWATH) ship has been developed from the latter concept, and a further refinement has been proposed in the use of the side struts to act as solid sidewalls for an air cushion under the deck. This typifies the semi-submerged air cushion vehicle (SSACV), in which the displacement hulls carry the major part of the weight and the air cushion such part as will optimize the operational parameters, including stability.

It has been suggested that the SSACV will be a fairly high-speed craft, with better economics than a sidewall hovercraft, but with a draught comparable with conventional shipping.

In addition, it may well be that the air cushion beneath the SSACV will provide a greatly enhanced icebreaking capability in the same manner as the air cushion icebreaking bow (ACIB).

In order to obtain a general assessment of the potential of the SSACV as a contender in improved Arctic transportation, a contract for a general feasibility study was placed by the Transportation Development Centre (TDC), with German & Milne Inc. (G & M), stipulating Dr. TKS Murthy as an outside SSACV consultant. Dr. Murthy is the holder of UK Patent No. 2010201 for an SSACV, icebreaking, liquid natural gas (LNG) or oil tanker.

2 SCOPE OF THE STUDY

To assess the potential of the SSACV icebreaking LNG tanker, it is necessary to compare the economics and performance of such a vessel with those of a conventional monohull ship (the Base Ship) performing the same task. This Base Ship will be an LNG tanker with a capacity for 140,000 m^3 of LNG. The LNG is assumed to be primarily methane, with a specific gravity (SG) of 0.43. Variations in SG are understood to be as great as 10%, but will not be considered.

For the purpose of the study it was agreed by TDC that the conditions might be taken as:

- Open Water - Sea State O, Calm
- Icebreaking - Uniform ice sheet, 7 ft (2.13 m) thick

Several bases of comparison were considered, but it was found impossible to adapt any particular scenario to permit a fair assessment. Finally, since only broad estimates were being made and, in some cases, no quantitative comparison was possible, it was decided to use a number of simple parameters, namely:

- Capital Cost
- Operating Cost
- Manning
- Speed attainable on 80,000 kW in open water
- Speed attainable on 80,000 kW in ice

Note: For the purpose of this study it will be assumed that propulsion power and fan power are derived from the same source in the SSACV tanker.

It must be appreciated that the environmental conditions being considered are ideal and do not represent those which would be encountered in practice. Consideration of practical aspects is covered in Section 6.

3 BASIS OF COSTS

The basis of costs will be the same for both vessels and has been estimated from data compiled and maintained by German & Milne Inc., which are considered proprietory and confidential. Details will be supplied on request.

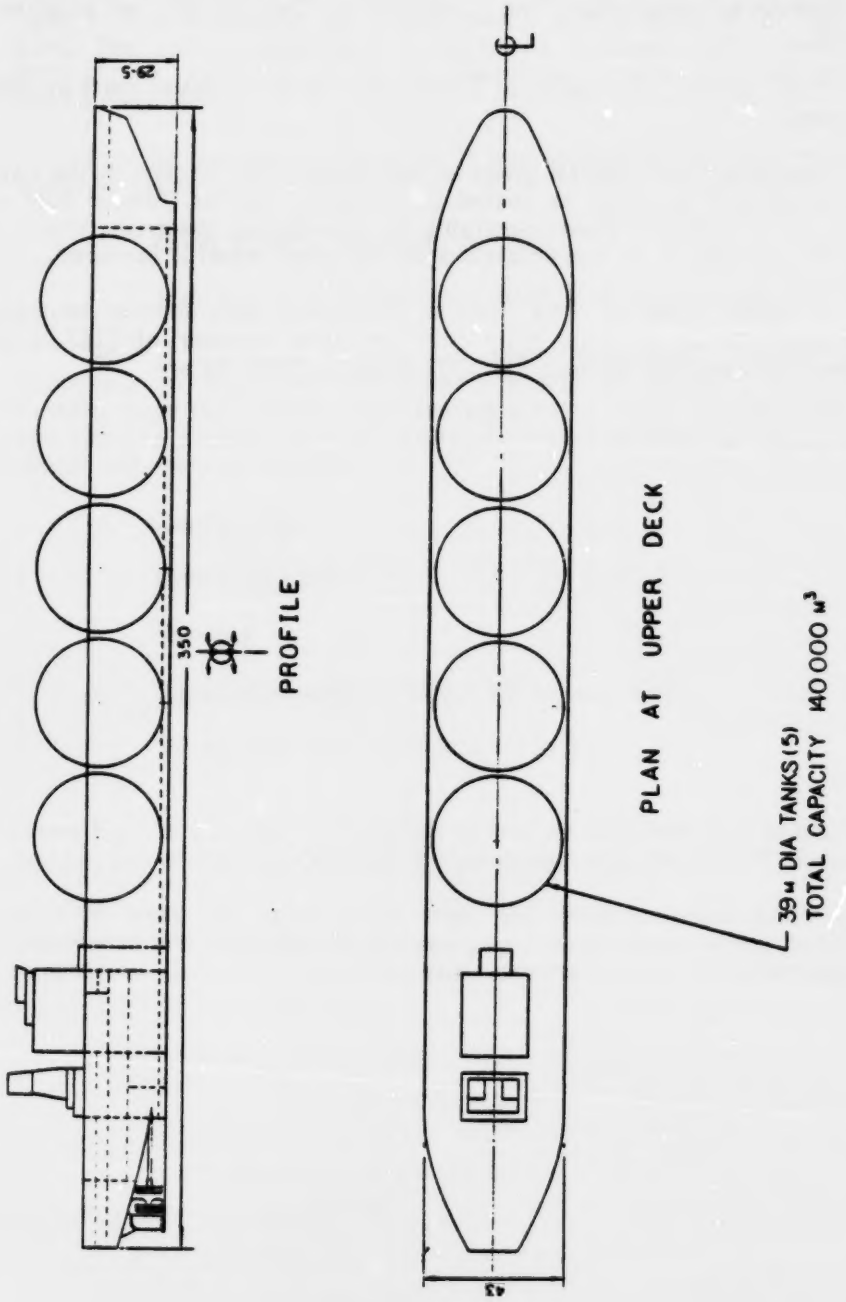
Fuel cost has been taken at $180/t, and other oil based costs at 50% of fuel cost.

Operating cost will be given as per annum cost, except in the case of fuel, which will be given as costed consumption figures. Annual fuel costs cannot be provided without specifying an operational spectrum, and it does not seem possible to do this unless one or the other vessel is favoured.

A recent study of crew costs in connection with another contract (C 3882) produced an average daily cost per crew member of $187.00 (1982 dollars). This may be taken as $170.00 per day in 1981 dollars

FIGURE 1: ICEBREAKING LNG TANKER (MONOHULL)

ALL DIMENSION IN METRES



4 THE BASE SHIP

4.1 DESCRIPTION

The Base Ship will be similar to the Arctic Pilot Project 140,000 m^3 LNG tanker designed to Ice Class 7. The layout of this vessel is shown in Figure 1, and dimensions and weights in Table 1.

TABLE 1: BASE SHIP, DIMENSIONS AND WEIGHTS

DIMENSIONS

Length WL	335.0 m
Beam	43.0 m
Depth	29.5 m
Draft	13.0 m
Displacement	140,000 t

WEIGHT ESTIMATE

Steel weight	40,500
Outfit	2,000
Machinery	4,600
LNG tanks	5,500
Lightship	52,600 t
Fuel	15,300
Crew	200
Cargo (SG LNG equals 0.43)	60,200
Ballast (icebreaking draft)	11,700
Displacement	140,000 t

4.2 POWERING

The resistance of the Base Ship in water has been estimated by conventional methods. The total resistance, the sum of frictional and residuary components,is translated into required shaft power by assuming a quasi-propulsive coefficient of 0.5. Powers were calculated for a range of speeds. The values given in Table 2 were obtained and are plotted in Figure 2.

TABLE 2 BASE SHIP, PERFORMANCE

1 OPEN WATER

Reference: Gertler, A Reanalysis of the Original Test Data for the Taylor Standard Series

$L = 335$ m S, wetted surface $= 17{,}949$ m^2

ehp $= \frac{1}{2}\rho S V^2 C_t$ $\Delta C = 0.4 \times 10^{-3}$

$V_k/\sqrt{L}$	V m/s	$R_e \times 10^{-9}$	$C_f \times 10^3$	$C_r \times 10^3$	$C_f + C_r + \Delta C = C_t \times 10^3$	P_e kW	Power kW
0.30	5.14	1.45	1.463	0.650	2.513	3,155	6,340
0.36	6.17	1.74	1.432	0.650	2.482	5,377	10,800
0.42	7.20	2.03	1.406	0.650	2.456	8,472	16,900
0.48	8.23	2.32	1.384	0.656	2.440	12,520	25,100
0.54	9.26	2.61	1.364	0.725	2.489	18,189	36,400
0.60	10.29	2.90	1.348	0.872	2.620	26,269	52,600
0.66	11.32	3.19	1.333	1.138	2.871	38,321	76,700

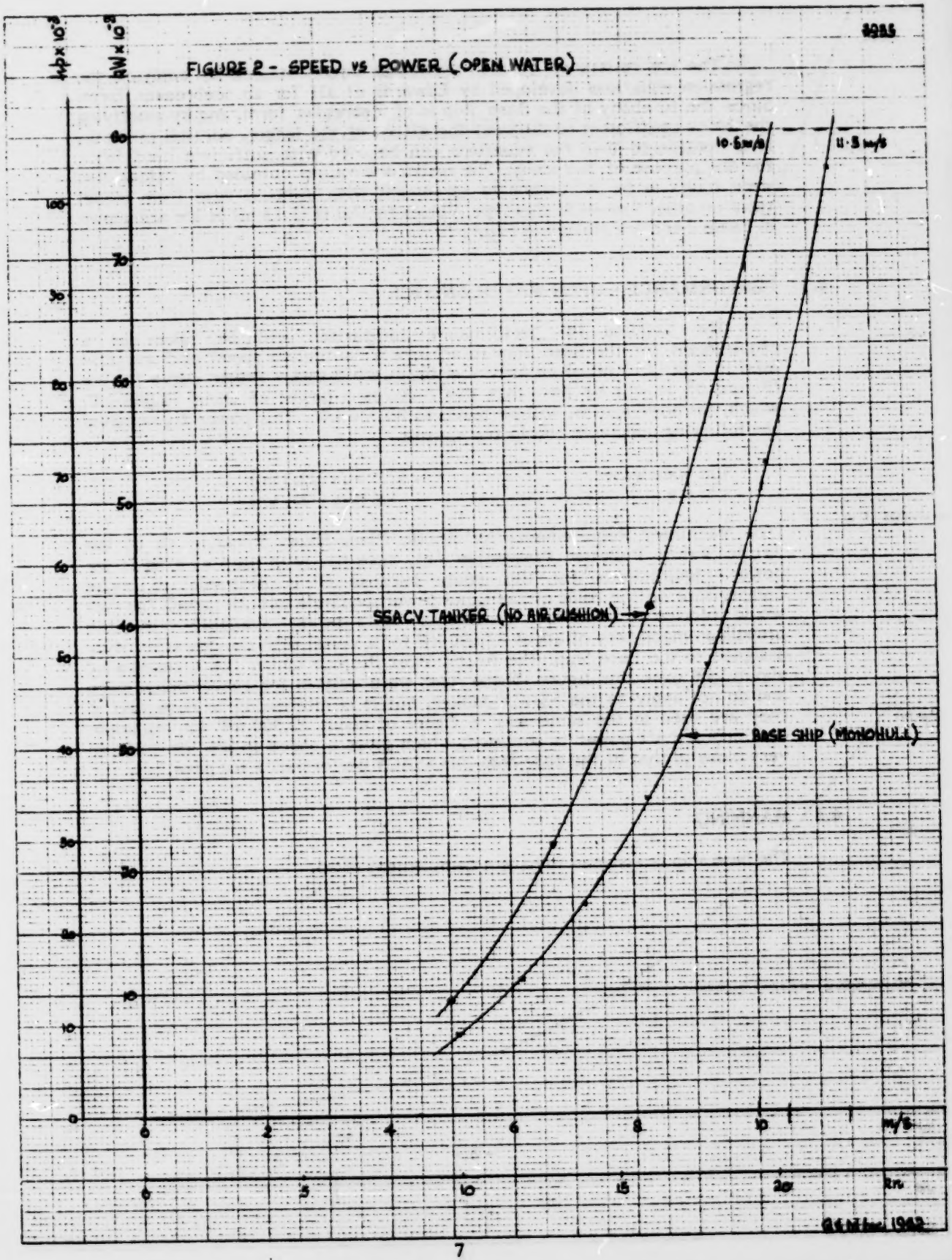
2 IN LEVEL UNBROKEN ICE

Reference: Edwards et al, Influence of Major Characteristics of Icebreaker Hulls on their Powering Requirements and Maneuverability in Ice

Ice resistance, $R = 600 + 44.4V$ lbs (V in ft/s)

Thrust delivered by 3 propellers, diameter 30 ft (9.1 m)

V ft/s	V m/s	R_{ice}	R_{water}	Req'd Thrust/shaft (+10% reduction)	Power shp	Power kW
0	0	600	0	220	56,992	42,500
4.92	1.5	819	5	302	104,196	77,700
9.94	3.0	1,037	21	388	167,329	124,800
14.75	4.5	1,256	47	478	247,667	184,700


3955
FIGURE 2 - SPEED vs POWER (OPEN WATER)
hp x 10⁻³
kW x 10⁻³
10·5 m/s
11·5 m/s
SSACV TANKER (NO AIR CUSHION)
BASE SHIP (MONOHULL)
m/s
kts

The ice resistance of this vessel has been obtained by applying the regression equations developed by Edwards et all for an icebreaker form. Since the forebody of the Base Ship is of icebreaker form, and by modifying the block coefficient to discount the effect of the longer, parallel midbody, it is considered that the equations can be used with sufficient confidence for the purpose of this study. The resistance values obtained by calculation were found to be in reasonable agreement with in-house data from model tests on other vessels of this type. Values for 7.0 ft (2.13 m) of ice are shown in Table 2 and are plotted in Figure 3.

4.3 PERFORMANCE - OPEN WATER AND ICE

The baseline for performance comparison will be taken as a requirement for the Base Ship to achieve a continuous speed of 3 kn (1.54 m/s) in 7 ft (2.13 m) of uniform ice. From the Speed v Power curve for ice (Figure 3), it will be seen that this requires a power of about 78,000 kW, which, for convenience, will be taken as 80,000 kW. This power figure gives the following maximum continuous speeds:

- In open water	- 11.5 m/s (22.3 kn)
- In ice	- 1.6 m/s (3.1 kn)

(The Machinery Requirements for Arctic Class Ships, Schedule VII of the Arctic Waters Pollution Prevention Act, have not been used as a basis for power requirements for this study because it is desirable to make comparisons on the basis of the same power available in both cases. The Act would require a minimum continuous shaft power available from the propulsion machinery of about 87,000 kW for a vessel with the characteristics assigned to the Base Ship. The Arctic Pilot Project tanker has a reserve of about 15% over this figure. These rules were written for conventional ships and do not apply to the SSACV tanker, but it is interesting to note that if they are used in the same way, a figure of about 22,300 kW for minimum continuous shaft power is obtained, which gives a speed in ice (no cushion) very close to that of the Base Ship (for SSACV performance see para 5.3).

4.4 MANNING

The base ship is assumed to have a complement of 45.

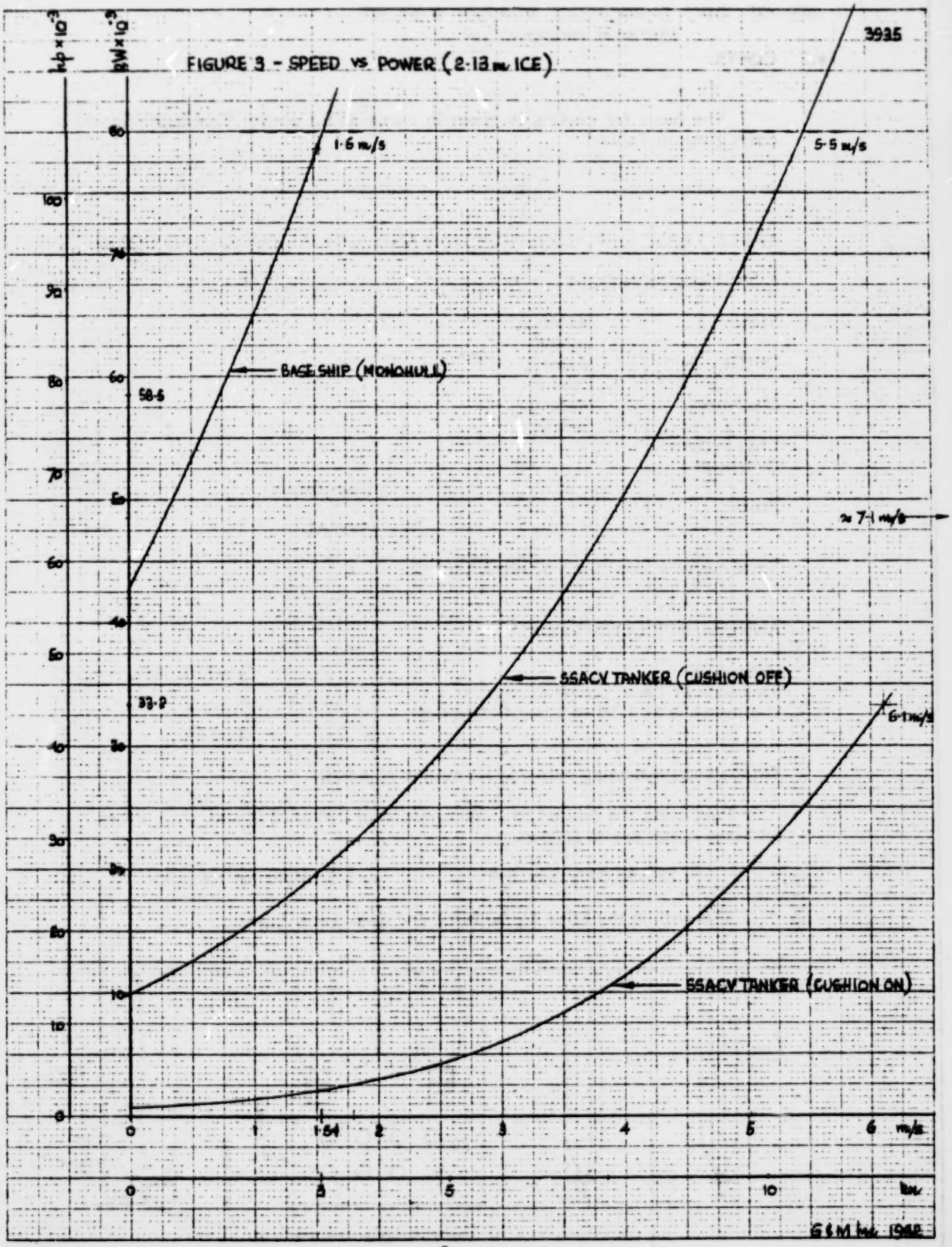


FIGURE 3 - SPEED vs POWER (2·13 m ICE)
hp × 10⁻³
BW × 10⁻³
3935
1·6 m/s
5·5 m/s
BASE SHIP (MONOHULL)
58·5
≈ 7·1 m/s
SSACV TANKER (CUSHION OFF)
33·2
6·1 m/s
SSACV TANKER (CUSHION ON)
m/s
km
G&M Inc. 1982

4.5 COSTS

The basis for costing is given in paragraph 3 above. This leads to the costs given in Table 3.

TABLE 3 BASE SHIP, CONSTRUCTION AND ANNUAL OPERATING COSTS

CONSTRUCTION COST			$\$x10^{-6}$
Steel	Material		23.29
	Labour		28.96
Outfit	Material		31.60
	Labour		12.48
Machinery	Material		70.00
	Labour		11.31
LNG System			52.00
Overhead			39.56
		Sub-Total	269.20
Profit			26.92
		TOTAL	296.12

Total construction cost - $296 million

ANNUAL OPERATING COSTS

Crew cost	45x170x365	2.79
Insurance at 1% of construction cost		2.96
Maintenance at 3.3% of construction cost		9.77
	TOTAL	15.52

Total annual operating cost - $15.5 million plus fuel.

Fuel costs are related to a consumption of 220 g/kW/h and, with other oil based costs, are as follows:

$220x80,000x180x10^{-6}$ = $ 3,168/hr (at 1.6 m/s)

This will be the same for both ice and open water cases.

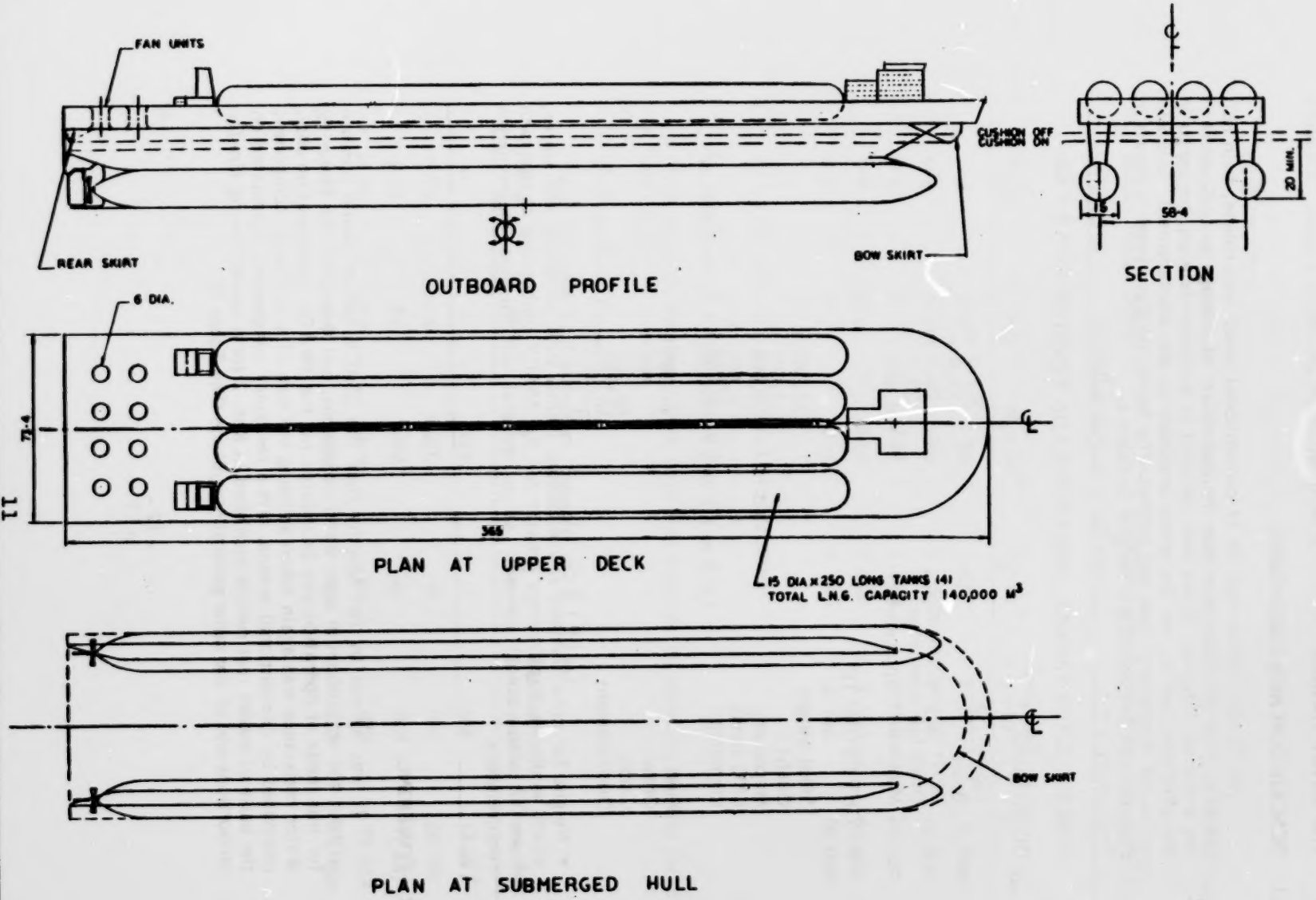


FIGURE 4: S.S.A.C.V. ICEBREAKING L.N.G. TANKER
ALL DIMENSIONS IN METRES

5 THE SSACV TANKER

5.1 DESCRIPTION AND DIMENSIONS

The SSACV tanker will be of conventional small waterplane, twin hull (SWATH) form. In order to reduce the probability of damage to LNG tanks in the event of collision, it has been decided to accommodate all the cargo in the platform, and to use the space available in the displacement hulls and struts for machinery, fuel and ballast. The layout of this vessel is shown in Figure 4 and dimensions and weights in Table 4.

TABLE 4 SSACV TANKER, DIMENSIONS AND WEIGHTS

DIMENSIONS

Length OA	365.0 m
Beam	73.4 m
Draft (air cushion off)	26.0 m
Draft (air cushion on)	20.0 m
Minimum Displacement	140,000 t

WEIGHT ESTIMATE

Steel weight	52,000 t
Outfit	3,000
Machinery	4,000
LNG tanks	6,000
Lightship	65,000 t
Fuel	14,300 t
Crew	500 *
Cargo	60,200
Displacement	140,000 t

* Weight for crew includes miscellaneous items, such as spares and stores, which would be significantly greater for the SSACV tanker due to the twin hulls (2 work areas) as well as to general air cushion maintenance requirements.

5.2 POWERING

It was agreed by Dr. Murthy that the SSACV tanker would operate without the air cushion in open water conditions, and therefore the powering for this mode of operation was calculated for the SWATH configuration only. Water resistance was again taken as being the sum of a frictional component computed by conventional means, and a residuary component, calculated on the basis of model test results published in Ref. 2 for a model having similar proportions and of the same general magnitude (See Table 5).

TABLE 5 SSACV TANKER, PERFORMANCE

1 OPEN WATER

Reference: Pien and Lee, Motion and Resistance of a Low-Waterplane Catamaran

Basis: Model 5266 $C_F = 0.0004$ (approximated for $V_K/\sqrt{L} < 0.7$)

$L = 320$ m S, wetted surface $= 37,600$ m^2 $\Delta C = 0.4 \times 10^{-3}$

ehp $= \frac{1}{2}\rho S V^3 C_T$

$V_K/\sqrt{L}$	V m/s	$R_e \times 10^{-9}$	$C_F \times 10^3$	$C_T + C_F + \Delta C = C_T \times 10^3$	P_e kW	Power kW
0.3	5.0	1.34	1.477	2.277	5,500	9,200
0.4	6.7	1.80	1.426	2.226	12,900	21,800
0.5	8.3	2.24	1.389	2.189	24,100	41,200
0.6	10.0	2.69	1.360	2.160	41,600	69,600
0.7	11.7	3.14	1.336	2.136	66,000	110,000

2 IN LEVEL UNBROKEN ICE (CUSHION OFF)

Ice resistance, $R = 195 + 14.5V$ tons (V in ft/s)

(This equation developed from Base Ship equation by adjusting linearly for the beam)

Strut width = 7.0 m

Thrust delivered by 2 propellers, diameter 30 ft (9.1 m)

V ft/s	V m/s	R_{ice}	R_{water}	Req'd thrust/shaft (+10% deduction)	Power hp	Power kW
0	0	195	0	108	13,067	9,750
4.92	1.5	266	10	152	26,138	19,500
9.84	3.0	338	40	208	47,637	35,500
14.75	4.5	409	90	275	79,362	59,200
19.67	6.0	480	160	352	123,400	92,000

3 IN BROKEN ICE (CUSHION OFF)

Reference: Decker, The Application of Air Cushion Configurations to Icebreaking

$$R/\rho_w g B h^2 = 1.16 + 1.23\, V^2/gh \quad \text{(using ore carrier forms)}$$

V ft/s	V m/s	R_{ice}	R_{water}	Req'd thrust/shaft (+10% deductions)	Power hp	Power kW
0	0	37	0	20	1,041	780
4.92	1.5	42	10	29	2,747	2,050
9.84	3.0	54	40	52	8,064	6,010
14.75	4.5	76	90	91	20,241	16,100
19.67	6.0	105	160	146	42,635	31,800

Ice resistance has been assumed to be a linear function of the beam, within the range of beams considered. Therefore, since the strut stems will have an efficient icebreaking shape, the resistance of each strut in ice may be assumed to be proportional to that of the Base Ship. This may appear to be somewhat optimistic, since the downward forces produced by small waterplane struts will be less than those produced by a conventional hull for the same change in draft. However, the flexural strength of ice is low, so that the vertical forces required to fracture the ice are relatively small.

Consideration of the SSACV tanker as configured in Dr. Murthy's Patent Application and in his notes on the subject led us to the conclusion that a bow skirt between the side struts will make no significant contribution to easing the passage of the vessel through the ice, because the struts will already be cutting their own paths before the air cushion icebreaking takes place.

It was suggested to Dr. Murthy at a meeting on 28 September 1981 that the contribution of an air cushion would only be effective if it was so arranged as to break the ice ahead of the struts. It was also suggested that, in the case of the tanker, passage through the ice was the requirement and that icebreaking per se was not. Consequently the provision of an air cushion around the strut stems only might be considered, since the hull arrangement is suitable and a substantial power reduction might be achieved. Dr. Murthy was not receptive to this idea because the reduction in cushion area would reduce the effectiveness of the application of a fore and aft rocking motion to the vessel by means of the air cushion.

As an alternative, extending the front skirt around the strut stems was suggested, and Dr. Murthy agreed that the full bow skirt should be configured to break ice ahead of the struts.

Unfortunately, with this 140,000 t vessel, skirts at bow and stern and a cushion pressure of 20.69 kPa (equivalent to 7 ft of ice), the change of draft from cushion off to cushion on will be about 10 m, which means that the submerged hulls will be only 1 m below the surface with cushion on. It is evident that this is not satisfactory, so some means of reducing the pressurized area of the hull must be found. Bringing the aft skirt forward or having separate cushions at bow and stern are possibilities, and ballasting could be considered. However, large quantities of ballast may reduce the buoyancy to an unacceptable degree in event of cushion failure. For the purpose of this assessment it is being assumed that a cushion on draft of 20 m minimum has been obtained.

With anything but a full length cushion, icing inside the tunnel due to spray could be a problem, but it is believed that suppression devices are being developed successfully, notably by the Aerojet Liquid Rocket Company, USA (RW Dyke at Fifteenth Canadian Symposium on Air Cushion Technology, Toronto, September 1981).

Further consultation on 2 October 1981 showed general agreement on the basis of the study and the work done to date, but Dr. Murthy was concerned that a skirt extended around the strut stems could be trapped between the ice and the stem and damaged, and insisted that the original configuration, front skirt between the struts, as shown in the patent application, should be used.

Our understanding of the mechanism of low speed air cushion icebreaking is that in practice the ice fails under or just behind the skirt. Consequently, in the configuration recommended by Dr. Murthy, the air cushion would simply break ice between the hulls. It would not contribute to easing the passage of the vessel through the ice, because the struts would have to break ice with their stems which must be ahead of the front skirt in order to provide an air seal.

Finally an arrangement for a semi-circular skirt with a radius compatible with joining the strut stems was proposed by Dr Murthy. With this form it is expected that ice will be broken ahead of the struts.

The power required for the air cushion (fan power) has been estimated by the British Hovercraft Corporation to be 62,800 hp (46,786 kW) (see Annex A). For their calculation an air gap, h, of about 4.5 in. was assumed. Comparison of this figure with data given in eference material, for example Ref. 3, indicates that this may be excessive. The problem here is that, since fan power is directly proportional to air gap, the power requirement is very dependent on the choice made from a limited data base covering a wide range of possibilities.

Informal discussion with a number of people involved in air cushion technology indicated clearly that the proposed SSACV tanker is well outside the present state of the art, and the word "art" is used advisedly. There is very little experience with solid sidewall ACV, and any that there is relates to high speed vehicles operating over water. BHC, for example, have no experience with solid sidewalls or with low speed icebreaking.

It seems evident that no really reliable estimate of cushion power can be made in the light of current experience, and consequently the BHC figure must be accepted as a first approximation.

Dr Murthy has rightly pointed out that, because of its very large beam, the SSACV will be capable of breaking 7 ft (2.13 m) of ice at cushion pressures as low as 200 lb/ft^2 (9.6 kPa), which would result in a considerable power saving. For a cushion pressure of 9.6 kPa the fan power required would be only about 21,500 kW as opposed to 46,786 kW required for 20.68 kPa (extrapolating from BHC data). Under the rules for the study this means that an additional 25,286 kW could be made available for propulsion, resulting in an increase in ice transitting speed from 6.1 to about 7.1 m/s (say, 16%), which is an appreciable gain.

However, in practice, the powering arrangement may not permit the transfer of power from fans to propulsion. It is more likely that separate powering would be provided, with propulsion power related to maximum ice transit speed and fan power related to some compromise between operational flexibility and economy. Any reduction in available fan power would affect adversely the pitch oscillating mode of ridged ice cutting suggested by Dr Murthy.

5.3 PERFORMANCE - OPEN WATER AND ICE

The performance of the SSACV icebreaking tanker is presented in Table 5 and plotted in Figures 2 and 3.

With 80,000 kW available, the SSACV tanker will be able to achieve the following maximum continuous speeds:

- In open water	-10.5 m/s (20.4 kn)
- In ice (cushion off)	- 5.5 m/s (10.7 kn)
(cushion on-19.4 kPa)	- 6.1 m/s (11.9 kn)
(cushion on- 9.6 kPa)	- 7.1 m/s (13.8 kn)

5.4 MANNING

BHC suggest that a labour force of four men will be required to maintain the skirts. This will be in addition to the normal ship's complement, and does not make any allowance for the additional engineering tasks associated with fans, fan drives and ducting, for which, very conservatively, two men will be added.

The SSACV tanker will thus require six men in addition to the forty five 'standard' crew required for the base ship. The total complement will be 51.

5.5 COSTS

Construction and annual operating costs are given in Table 6.

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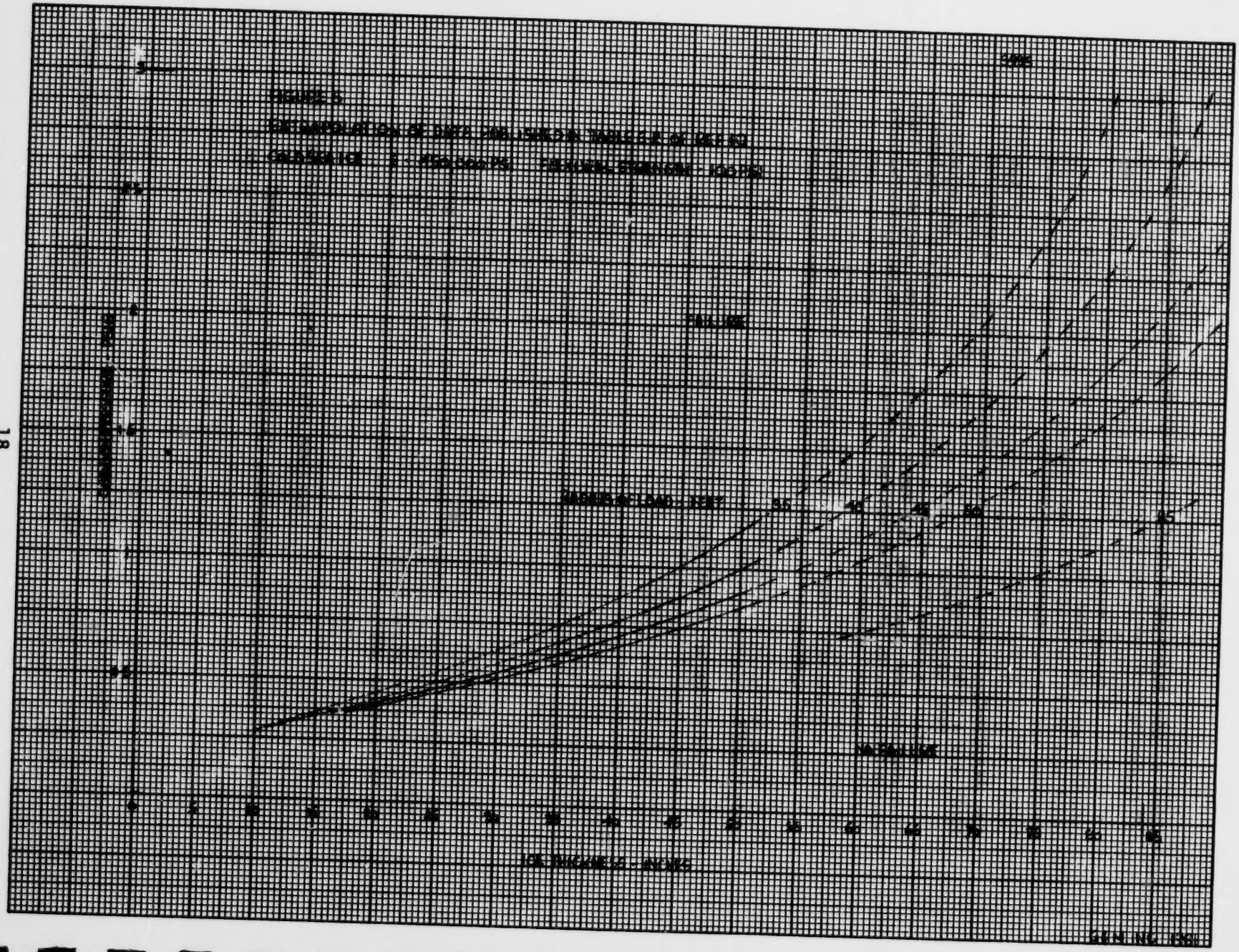

FIGURE 5

EXTRAPOLATION OF DATA PUBLISHED IN TABLE 5.2 OF REF 10

COLD SEA ICE E = 450,000 PSI FLEXURAL STRENGTH - 100 PSI

TABLE 6: SSACV TANKER, CONSTRUCTION AND ANNUAL OPERATING COSTS

CONSTRUCTION COST

			$x10-6
Steel	Material		29.90
	Labour		37.18
Outfit	Material		47.40
	Labour		18.72
Machinery	Material		49.00
	Labour		7.92
LNG System			52.00
Overhead			47.87
		Sub-Total	289.99
Profit			29.00
		TOTAL	318.99

Total construction cost (SWATH ship) - $319 million.

The cost of including the air cushion for the purpose of icebreaking is estimated separately below. For this purpose reliance has been placed on the costs supplied to Dr. Murthy by BHC.

Lift fans 8 @ 4,290 lb @ $(60 x 2.2)		4.53
Machinery (estimate for an electrical fan powering system)		5.00
Bow skirt 42,000 lb @ $(12 x 2.2)		1.11
Rear skirt 29,000 lb @ (12 x 2.2)		0.77
Skirt deployment and retraction equipment	Say	2.00
Ducting	Say	1.00
Control System	Say	2.00
	TOTAL	16.41

Total construction cost (SSACV tanker) - $335 million.

ANNUAL OPERATING COSTS:

Crew cost	51x170x365	3.16
Insurance at 1% of construction cost		3.36
Maintenance at 3.3% of construction cost		11.09
	TOTAL	17.61

Total annual operating cost - $17.6 million plus fuel

Fuel costs are calculated on the same basis as those given in paragraph 4.6 and, since the same power (80,000kW) is used, costs will be the same.

6 OPERATIONAL ASPECTS

6.1 THE BASE SHIP

Conventional icebreaking by monohull ships is a fairly straightforward operation which has been practiced for many years. However, the design has been largely a matter of judgement, experience and inituition, and the application of "good practice". The increasing interest in Arctic transportation has led to more serious study of the problem, and changes in stem shape and ship lines, together with the use of accessories, such as bubbler systems and low friction finishes, have contributed to more efficient vessels.

The Base Ship selected for this study faces few unknowns, which is why steel, monohull ships are being used exclusively at this early stage of Arctic development.

6.2 THE SSACV TANKER

Before discussing the operational aspects of the SSACV tanker, it should be made clear that an icebreaker is a vessel which is designed to break ice and to leave a channel through which following ships, which do not necessarily have an icebreaking capability, may pass. A vessel which is designed to transit ice on its own does not have to leave a channel behind, although this will certainly be the case with any monohull ship. The SSACV tanker, which would operate alone, is only required to transit ice, and only needs to break whatever may be required for the passage of its twin struts.

The forward skirt of the SSACV tanker is assumed to have a semicircular planform when deployed, the ends being attached essentially in line with the leading edge of the side struts. It is claimed by Dr. Murthy that with this configuration ice will be broken in front of the struts. At present, in the absence of any experience or testing with a semicircular cushion, this is speculation, but may well prove to be correct.

This comparative study only considers a uniform icefield, and no account has been taken of the pactical problems which would be encountered in polar icefields with ridges and snow cover. The practical aspects of operating an air cushion vehicle the size of a tanker can only be related to the small scale trials which have been conducted to date, and these have certainly indicated that improvements are required in skirt materials to improve flexibility and life at low temperatures and in spray suppression to reduce icing problems. To be effective, the skirt and its operating mechanism must be capable of trouble free operation through a round trip voyage since major maintenance en route will be virtually impossible and, in any case, delay would defeat the purpose of the air cushion installation.

In calm open water, the performance of the SSACV tanker is inferior to that of the Base Ship. However in rough water the difference will not be so great and may well favour the twin hulled ship. Provision will have to be made to cover or otherwise protect the skirts and operating mechanism when retracted, as well as the fan exits.

A major problem with both the Base Ship and the SSACV tanker would be turning in ice. It is not known how this problem is being addressed for Arctic Pilot Project vessels, but the assistance of an icebreaker could well be needed.

7 SUMMARY

Table 1 and Figure 6 compare estimated quantitative data appropriate to the Base Ship, a 140,000t monohull, and an SSACV tanker of the same capacity. Both have Class 7 icebreaking capability.

TABLE 7: COMPARATIVE COST AND PERFORMANCE DATA

Item		Base Ship	SSACV Tanker	Gain/Loss
Construction Cost	M$	296	335	+ 39.0 (13%)
Total Installed Power	kW	80,000		Nil
Annual Operating Cost (less fuel)	M$	15.5	17.6	+ 2.1 (13%)
Manning		45	51	+ 6.0 (13%)
Fuel Cost per hour	$	3,168	3,168	Nil
Speed in Open Water	m/s	11.5	10.5	- 1.0 (9%)
Speed in Ice	m/s	1.6		
-cushion off			5.5	+ 3.9 (243%)
-cushion on (19.4 kPa)			6.1	+ 4.5 (281%)
-cushion on (9.6 kPa)			7.1	+ 5.5 (344%)

The cost differences are not inordinate, nor is the margin in open water performance. However the gain in ice transiting performance is dramatic.

The SSACV tanker requires a high proportion of the available power for operation of the air cushion, which is a complicated, delicate and expensive system. In open water, the air cushion is not used because it would reduce the power available for propulsion and increase resistance with consequently reduced performance. Its use would also increase maintenance. In ice, with cushion on (19.4 kPa), the total power available for propulsion is reduced to 33,214 kW and the transit speed would be 6.1 m/s, which is only 11% greater than the speed with cushion off. At a reduced cushion pressure (9.6 kPa) less power is required for the cushion, making 58,500 kW available for propulsion. This raises the transit speed to 7.1 m/s, a gain of 29% over the cushion off case.

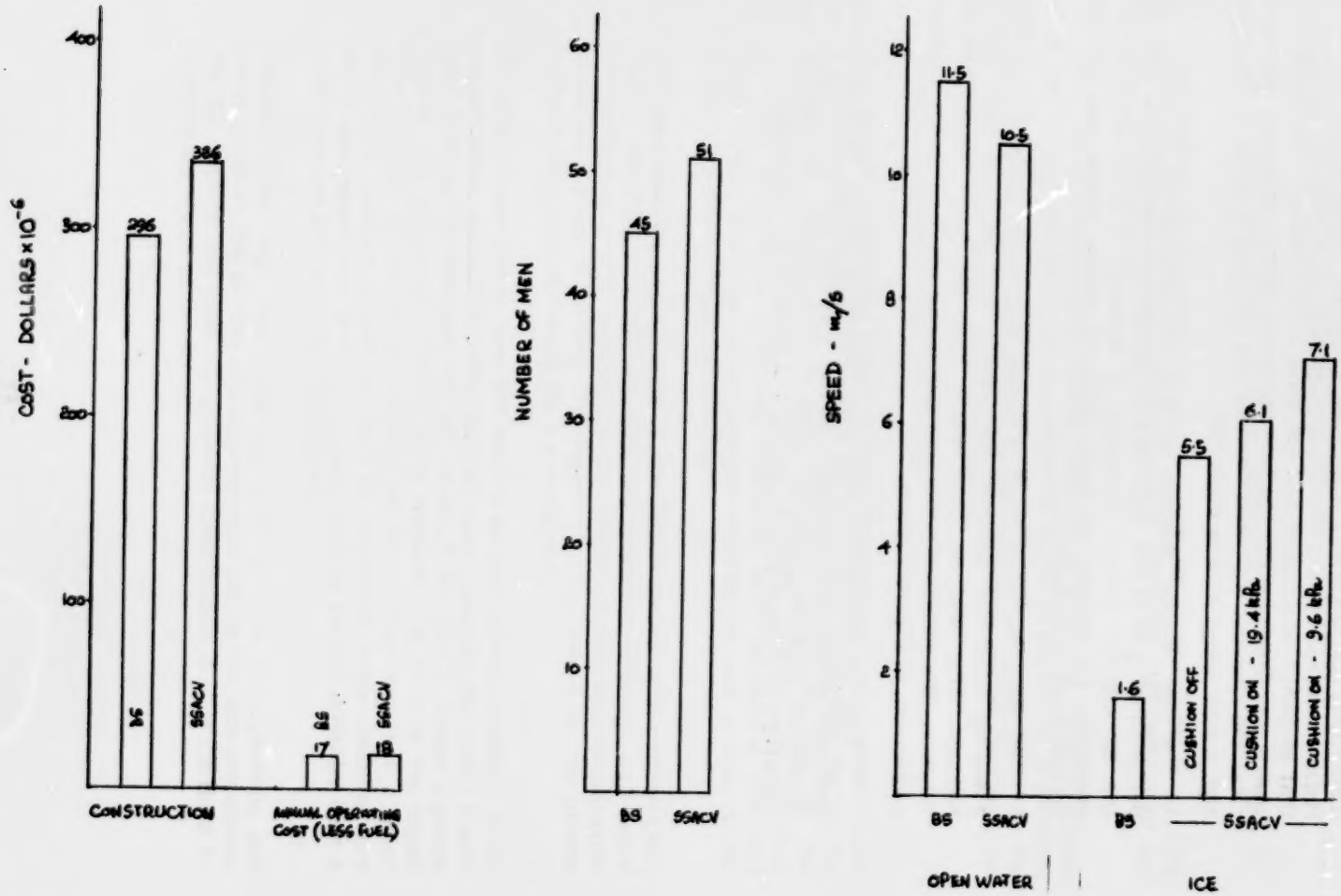

FIGURE 6 - COMPARISON BAR GRAPHS

In the icebreaking role, that is to say when regarding the SSACV simply as an icebreaking vessel, it may well have advantages over a conventional icebreaker, but the effect of cushion lift on the vessel and the cushion power requirement will require careful consideration. It should also be borne in mind that an SSACV icebreaker will not leave a clear passage, but simply a path of broken ice. Consequently the next following vessel, which will be required to clear the channel, will need considerable ice strengthening if the potentially high transit speed is to be exploited.

If full, bubble pressure is used for icebreaking, it seems that the performance gain is not sufficient to justify the complication and costs for the air cushion system. However, if breaking can be achieved at lower pressures, the concept at least starts to appear viable, but the practical problems are formidable.

It should be noted that, if higher open water speed is demanded, the powering difference between the SWATH ship and the Base Ship will be reduced. This will be particularly true for rough water performance. However, the increased power will also favour the SSAV in ice, cushion off or cushion on, as Figure 3 shows.

Also, in connection with low cushion presures, there is probably an optimum relationship between cushion pressure and strut separation (skirt diameter). However, reserve power will always be required to cater for variations in cushion flow caused by rough ice or skirt damage.

8 CONCLUSIONS

The Base Ship (monohull) is approximately 10% faster than the SSACV icebreaking tanker in calm open water.

The SSACV icebreaking tanker is approximately 290% faster than the Base Ship in uniform, level ice.

The higher performance in ice is achieved at the expense of 13% increase in construction cost, 13% increase in operating cost (less fuel) and 13% increase in manning.

The SSACV icebreaking tanker requires 67% of the total available power for the air cushion. If all the available power is applied to propulsion, almost the same performance can be achieved in ice without the complication and expense of an air cushion. However, in this configuration the tanker would not break a channel in the ice.

The following conclusions are drawn:

1 The SSACV icebreaking tanker offers a considerable improvement in ice transiting performance in smooth ice of uniform thickness, but at the expense of higher capital and operating costs, as well as the complications introduced by the air cushion system. The practical problems involved in designing, operating and maintaining this very large SSACV with the characteristics required for use in real Arctic winter conditions will be formidable.

2 The ice transiting performance of the SSACV tanker with cushion OFF, that is operating as a SWATH ship, is impressive. Although well below the performance with cushion ON, the additional cost and complication of the air cushion is lacking. This suggests a requirement for the further study of multi-hulled vessels for ice transiting.

3 The LNG tanker per se does not require an icebreaking capability in the proper sense, because it needs only to make a passage for itself. Power used to break ice between the side struts of the SSACV tanker is power wasted, the full length cushion being required only to rock the vessel fore and aft in special circumstances. It is submitted that a much smaller air cushion around the stem of each strut would consume very much less power and that the vessel could be rocked laterally (and a little fore and aft) with the same effect.

Simplicity and high ice transiting speed are the requirements, and power saved on the air cushion is power that can be put into propulsion. The air cushion will make its greatest contribution if it is confined to breaking ice ahead of the struts.

4 Until some testing is carried out it is not possible to say whether the SSACV icebreaking tanker is a viable proposition, although its performance in level ice of uniform thickness is likely to be markedly superior to that of an equivalent monohull ship for the same total power. The added cost and complexity of the air cushion might be justified if the air cushion system could be demonstratively simple and reliable.

5 The SSACV appears to have potential as a high speed icebreaker, but does not provide a clear channel for following ships. There could be a problem in avoiding excessive draft change at high cushion pressures, and the use of power for the air cushion, as opposed to adding it to the basic propulsion power, must be carefully considered and related to its complication and cost.

9 RECOMMENDATIONS

The following recommendations are made:

1 That a further, brief study should be made comparing an SSACV in the pure icebreaking role with a conventional icebreaker of the same class.

2 That, following the study, a series of model tests should be conducted to confirm the open water and icebreaking performance estimates of the SSACV icebreaker in the following configurations:

- cushion off
- cushion on, full length cushion
- cushion on, cushion around side strut stems only.

These tests should be made at several cushion pressure levels.

The scale of the model should be as large as possible.

3 That, following successful model tests, a small vessel should be designed, in the configuration of the most satisfactory model, with a view to construction and full scale concept trials.

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ADDENDUM 1

It was requested by TDC that the effect of halving the fan power required might be considered (assuming that BHC had overestimated by 100% the fan power required).

Since any overestimate could only be related to the gap height (flow), the changes would be as follows:

1 Cost The number of fans would be halved, resulting in a saving of about $6 million on construction cost (1.8%) and $0.31 million per annum on maintenance (1.6%).

2 Performance The performance in open water would be unchanged, but the power available for propulsion in ice would be increased to 46,605 kW, with a resulting speed increase to about 6.8 m/s (2%).

It should be noted that halving the number of fans will have a considerable bearing on the rate at which the vessel may be rocked in the icebreaking mode suggested by Dr. Murthy.

ADDENDUM 2

Without benefit of model or full scale testing, the assessment of the performance of a novel design is dependent on the choice of method and the selection of appropriate constants. This is largely a matter of opinion based, hopefully, on experience, but differences in results are inevitable, and the SSACV tanker is no exception. There is sufficient difference between the calculations by German and Milne (Main Report) and Dr. Murthy (Annex C) to illustrate this point, and a comparison has been made of the resistances in ice obtained by the various methods chosen. Differences in basic conditions are noted.

1	2	3	4	5	6	7	8	9
0	600		157	244	171			
1		1548				766	273	517
2								
3	827	1652	224	343	240	808	290	555
4								
5		1860				892	325	631
6	1058		311	462	323	950	350	684

1	Speed - kn			
2	R.tons.	Base Ship G&M	See page 6	
3		Base Ship TKSM	Lewis/Edwards equation	
4		SSACV G&M	5.60 m strut - no cushion	
5		SSACV G&M	8.75 m strut - no cushion	
6		SSACV G&M	8.75 m strut - cushion on (30% reduction)	
7		SSACV TKSM	8.30 m strut - no cushion	
8		SSACV TKSM	6.70 m strut - cushion on	- 405 psf
9		SSACV TKSM	8.00 m strut - cushion on	- 100 psf

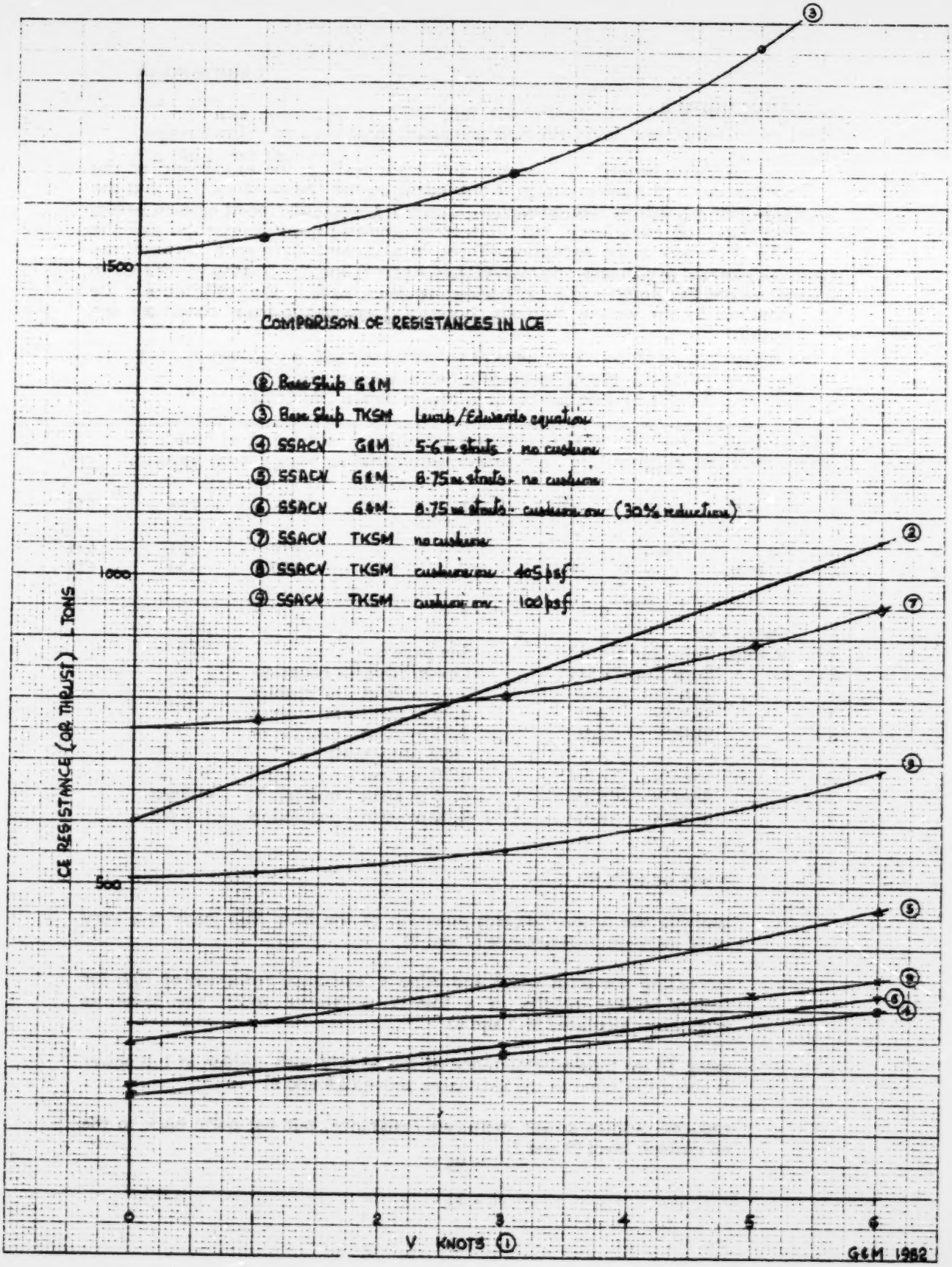
It will be seen that:

- applying the Lewis/Edwards equation to the Base Ship leads to a very large increase in resistance (2 and 3)

- increasing the strut width and using a resistance reduction of 30% for cushion effect results in a SSACV resistance much the same as originally calculated (4,5 and 6)

- using TKSM figures, resistance of the SSACV with cushion at full power is about the same as the estimates above (8,4 and 6)

- reduced cushion power increases resistance, but not quite back to the no cushion level (9 and 7)


COMPARISON OF RESISTANCES IN ICE
② Base Ship G&M
③ Base Ship TKSM Lewis/Edwards equation
④ SSACV G&M 5-6 m struts, no cushion
⑤ SSACV G&M 8.75 m struts, no cushion
⑥ SSACV G&M 8.75 m struts, cushion on (30% reduction)
⑦ SSACV TKSM no cushion
⑧ SSACV TKSM cushion on 405 psf
⑨ SSACV TKSM cushion on 100 psf
ICE RESISTANCE (OR THRUST) - TONS
1500
1000
500
0
1
2
3
4
5
6
V KNOTS ①
G&M 1982

ANNEX A

Letters from the British Hovercraft Corporation Limited
to Dr.TKS Murthy

TFA/PC/2776, dated 11 September 1981
TFA/PC/2778, dated 24 September 1981

BHC

british hovercraft corporation limited

[redacted] TELEX 86131 TELEGRAMS BRITHOVER COWES TELEX
HOVERPROD COWES TELEX

EAST COWES ISLE OF WIGHT PO32 6RH ENGLAND

Telephone 29412 Ext. No. 26
Our Ref TFA/PC/2776
Your Ref
Date 11th September, 1981.

Dr. Murthy,
4, Birchington Close,
Vinter's Park,
MAIDSTONE.
ME14 5PF

Dear Dr. Murthy,

The Ice-Breaking SSACV

I have pleasure in forwarding herewtih two copies of our note Ref. TFA/ANK/PC/2769 prepared in response to your verbal request and subsequent written order dated 8 September 1981.

Yours sincerely,
BRITISH HOVERCRAFT CORPORATION LTD.

T.F. ARLOTTE
Chief Dynamics Engineer

Enc.

BRITISH HOVERCRAFT CORPORATION LIMITED IS A SUBSIDIARY OF WESTLAND AIRCRAFT LIMITED
REG No 3[illegible] ENGLAND REGISTERED OFFICE WESTLAND WORKS YEOVIL SOMERSET

BRITISH HOVERCRAFT CORPORATION LTD., OSBORNE, ISLE OF WIGHT.

The Ice-Breaking SSACV - Some Initial Assessments of Cushion Power and Skirt Geometry Requirements

INTRODUCTION

The following notes provide our initial assessment of some of the design aspects associated with the cushion support system. The item numbers are those of the list tabled at our meeting of 26 August 1981 (Appendix 1 herewith).

MACHINERY TO PROVIDE LIFT POWER (item 1)

(a) Cushion Volume Flow

Based on the geometry of the 300m length x 50m width cushion at pc = 405 p.s.f., and taking a C_Q = 0.001, the cushion flow would be 94,200 cusecs. We believe that this is excessive and that a more realistic method would be to base the estimate on an equivalent air gap below the seals. This could be based on the SR.N4 Mk.1 for which Q = 20,000 cusecs, pc = 53.8 p.s.f. and cushion perimeter = 334 ft. The EAG is then about 4.5 inches. Using this value and allowing an extra 5% for lubrication the calculated flow is 58,000 cusecs.

(b) Fan Head

Taking the proposed relation between fan head and cushion pressure as H/Pc = 1.25, the fan head is 506 p.s.f.

(c) Size and Number of Fans

This head and flow can be supplied by 8 centrifugal fans of 16.25 ft. diameter, rotating at a speed of 645 r.p.m. The tip speed of the fans would be 550 ft./sec. This is within the strength limit for fans made of medium strength aluminium alloy.

(d) Fan Power

For an assumed fan efficiency of 0.85 the fan power required is:-

$$HP_F = \frac{H \quad Q}{550\,\eta} = \frac{506 \times 58000}{550 \times .85} = 62{,}800 \text{ H.P.}$$

(e) Fan Weight

The weight of each impeller, from BHC experience, would be 4290 lb. The total weight of the impellers would therefore be 34,300 lb.

SKIRT GEOMETRY (items 5 and 10)

(a) Bow Skirt

As agreed, we have evaluated the shape of a bow skirt that is 10m deep with a 2m deep finger.

(a) cont./ .....

A section that meets these requirements is shown on figure 1. The skirt consists of an outer material loop, 41 fingers of 4ft. width and inner cables. It should be noted that there are a number of possible skirt shapes that could be used. These would result from variations in the attachment locations and finger geometry. A more extensive study would be needed to examine these other skirt shapes.

It will be seen that the inner attachment of the cables is led to a point 1m above the wet deck. This is to allow for machinery to reduce the skirt depth by shortening the cables.

The modified shape of the bow skirt with a shortening of the cables is shown on figure 2. Here the length of the cable has been reduced progressively from 95 ft. (design value) to 74 ft. The geometry and loads for these sections are tabulated below.

cable length (C1) ft.	95	85	80	78	76	74
skirt depth ft.	36.1	31.75	28.3	26.6	24.65	22.35
skirt span ft.	95.75	86.57	82.41	80.83	79.31	77.85
outer loop tension, lb./ft.	7,600	9,987	12,100	13,400	15,230	17,950
cable load, lb./cable	41,000	50,400	59,100	64,300	71,480	82,360

Notes:

Skirt Depth is the distance, normal to the wet deck, from the inner attachment to the finger tip.

Skirt Span is the distance, parallel to the wet deck from the inner attachment to the cushion datum at the finger tip.

Cable load is based on the cables being at finger width (4 ft.) apart.

Finger geometry is represented (on figures 1 and 2) by the outer face line and a line joining the tip to the finger tail. The symbols (+, etc.) are the cushion datum points.

With a shortening of the cables, the loads in the outer loop and cables are significantly increased. The foregoing estimates refer to depth change of the inflated skirt. Powering requirements will depend upon the depth change, and rate of change required. Retraction loads for the non-inflated skirt would merely involve the weight of the system.

cont./ .....

For the outer loop of the bow skirt it would be necessary to use multiple layers of skirt material. Our current estimate of the total skirt material weight (for loop and fingers) is 500 oz/sq. yard. Thus, the bow skirt weight would be about 42,000 lb., excluding cables.

It may be noted that the outer loop strength requirement to deal with the cable length reduction to 74 ft. leads to the use of about 4 layers of present materials (30 oz/yd^2 nylon cloth coated with neoprene). This requirement may imply an insufficiently flexible loop and can be reduced by either using a double-loop configuration or by reducing the maximum depth change.

(b) Rear Skirt

A possible design for a rear skirt is shown on figure 2. This design, which is based on an experimental fit to an SR.N6 craft, would require some air to be directed into it to give a slight pressure differential above cushion pressure.

The depth of this skirt could be adjusted by having cables attached to the lowest point.

Assuming a material weight of 200 oz/sq.yd. the total rear skirt would weigh about 29,000 lb., excluding cables.

MANNING REQUIREMENTS SPECIFICALLY RELATED TO THE AIR CUSHION SYSTEM (item 8)

It is estimated that a labour force of 4 men would probably be required to maintain the skirts. This is similar to that required for an SR.N4 for a similar peripheral length of skirt and a similar utilisation. Methods of handling and handling equipment requirements would need to be considered in detail.

COST ASPECTS (item 9)

The design is not sufficiently detailed at this stage to permit a complete costing of the air cushion system machinery and equipment.

For approximate budgetary purposes it may be noted that typical costs, at SR.N4 size, are £10-12/lb for skirts and £55-60/lb for fans.

SKETCH OF G.A. (items 11, 2 and 3)

A general scheme of a layout for the fans and skirts is shown on figure 4.

cont./ .....

FURTHER ASPECTS

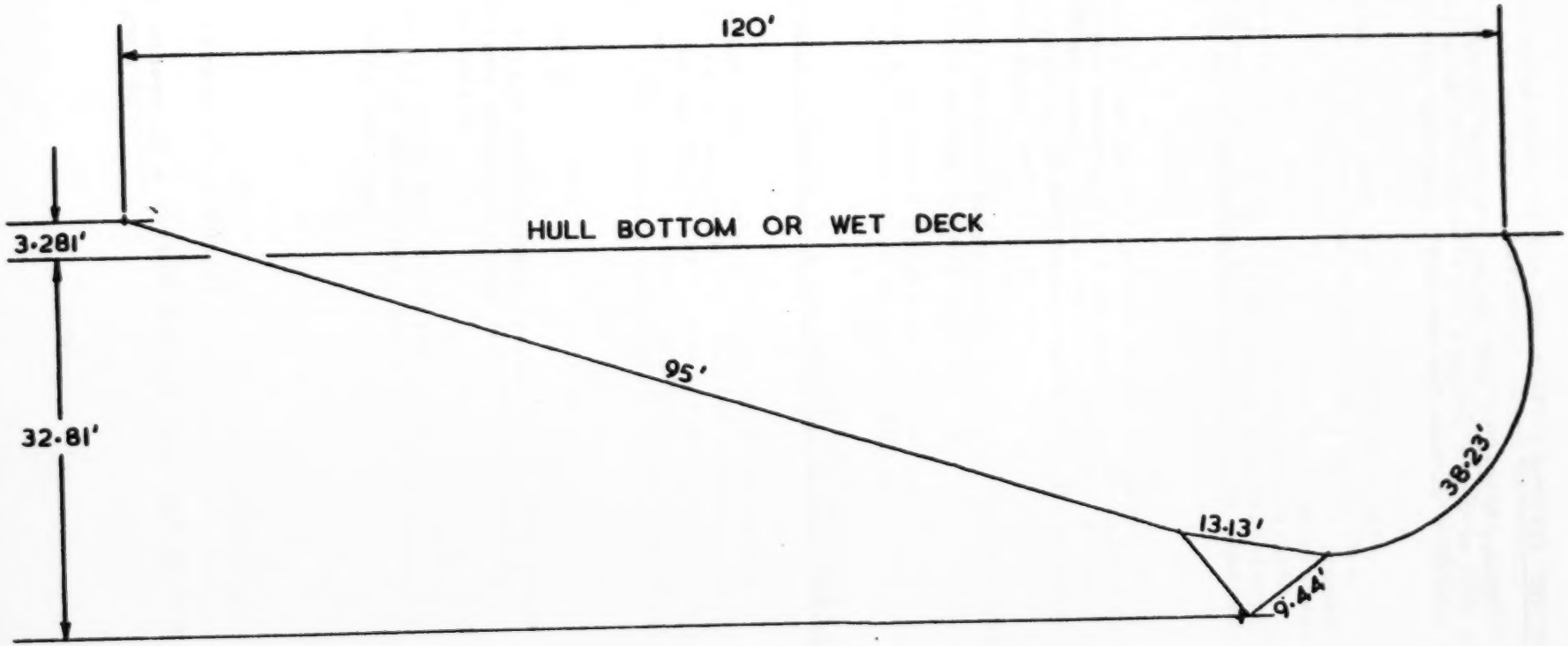
Items 6 and 7, and dynamic motion aspects of item 10, are topics which require more extensive detailed study than is possible within the present, limited, initial review.

Written by .....A.N. Key.
A.N. KEY

Approved by .....T.F. Arlotte
T.F. ARLOTTE

Ref: TFA/ANK/PC/2769
Date: 8th September, 1981.

P 62 . SSACV BOW SEAL (INNER ATTACHMENT 1 METRE ABOVE WET DECK)



SCALE = 1 / 150

FINGER WIDTH = 4'

FIG. 1

	OUTER LOOP LENGTH, Ft	CABLE LENGTHS, Ft
P 67 ▲	38.2298	74.0000
P 66 ▼	38.2298	76.0000
P 65 ■	38.2298	78.0000
P 64 ×	38.2298	80.0000
P 63 ●	38.2298	85.0000
P 62 +	SSACV BOW SEAL (INNER ATTACHMENT 1 METRE ABOVE WET DECK)	

VARIATION OF INFLATED SHAPE BY CABLE SHORTENING.

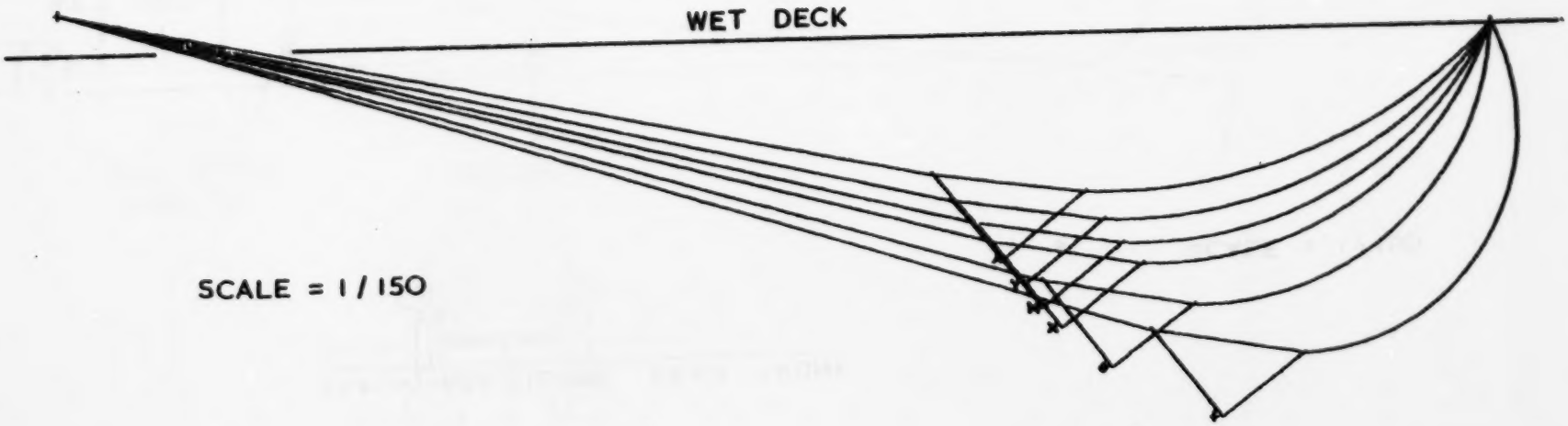


FIG. 2

SSACV MULTILOBE REAR TRUNK

SCALE = 1/150

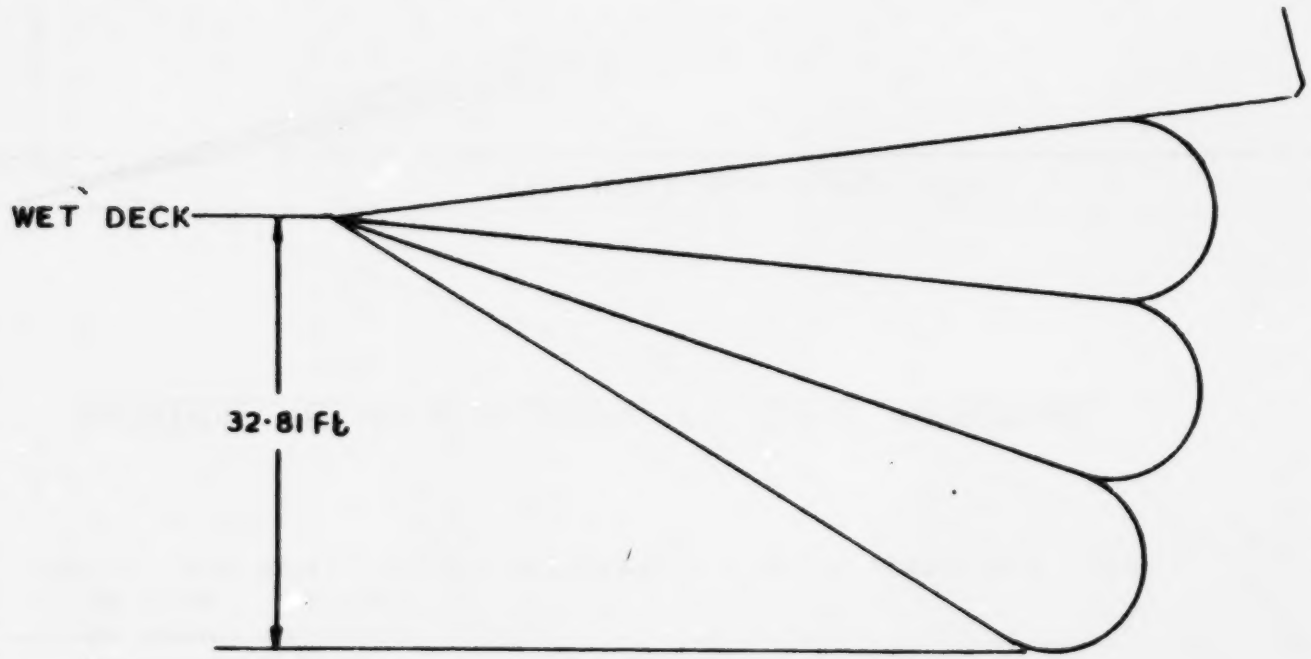


FIG. 3

GA. SKETCH SHOWING AIR CUSHION SYSTEM.

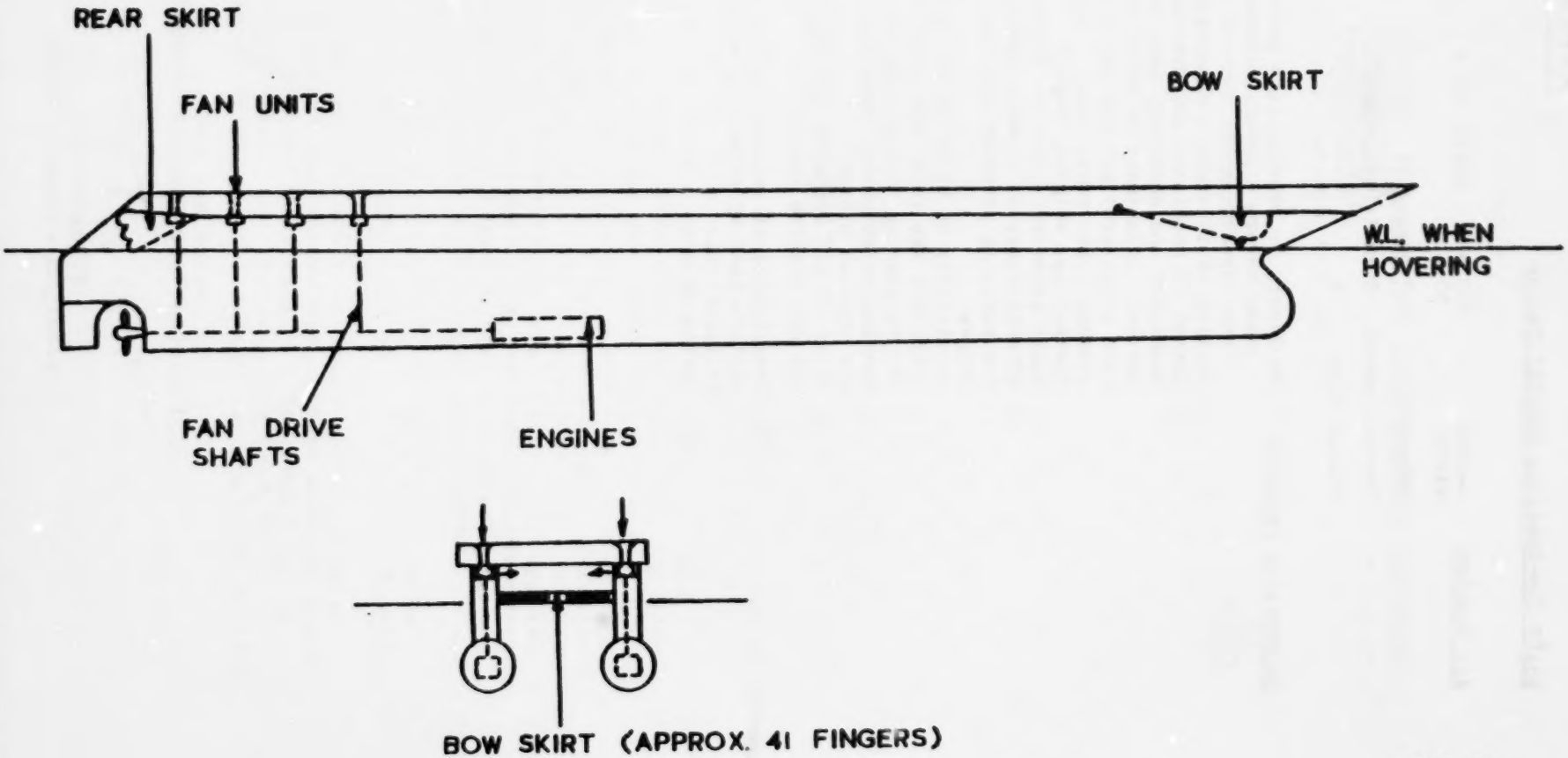


FIG. 4

APPENDIX 1

SSACV Ice-Breaking LNG/Oil Tanker

Air Cushion	Length	300 m	Depth 10 m
	Width	50 m	
	Pressure	2.8 psi	
	Forward speed	8 knots(maximum)	
	Volume flow	?	

Information required

1. Machinery to provide lift power - prime movers - power, dimensions, weights; fans; transmission; control system.
2. Ducting reqirements - dimensions, weight.
3. Layout of machinery and ducting - location of units.
4. Machinery and machinery arrangement for skirt control (adjustment of depth of hemline below hard structure) and for skirt retraction and stowage (open water case) - space requirements, dimensions, weights, layout.
5. Skirt geometry - type of segments, weight, material
6. Effectiveness of blowout doors in the side struts for reducing pressure fluctuations caused by ice ridges.
7. Possibility of bleeding air from the cushion to lubricate the inner and outer surfaces of the side struts passing through the ice layer.
8. Manning requirements specifically related to the air cushion system.
9. Costs of all machnery and equipment required for the air cushion system.
10. Feasibility of inducing pitch and heave (roll ?) oscillations by adjusting skirt depth - period of oscillations.
11. Sketch of G.A.

BHC

British Hovercraft Corporation Limited

[redacted] TELEX 86761 2 TELEGRAMS HOVERPROD COWES

EAST COWES ISLE OF WIGHT PO32 6RH ENGLAND

Telephone Ext. No. 26
Our Ref. TFA/PC/2778
Your Ref.
Date 24th September, 1981.

Dr. T.K.S. Murthy,
4, Birchington Close,
Vinter's Park,
MAIDSTONE,
Kent.
ME14 5PF

Dear Dr. Murthy,

The Ice-Breaking SSACV

The following additional points are made in amplification of our note Ref. TFA/ANK/PC/2769:-

(1) The basic bow skirt shape given in Figure 1 and the skirt shapes, tensions and cable loads given in Figure 2 and on page 2 were all obtained using our extensive skirt design computer program BH.1489. The part of the program used for this work provides skirt section shapes and tensions for a wide range of configurations. Figures 1 and 2 are copies of the actual plotter print out for the SSACV initial proposed arrangement. The program can also be used to examine skirt behaviour under externally applied contact loads. Further programs complete our present skirt Computer Aided Design system by providing the developed shapes for skirt panels and fingers.

(2) For full bow skirt retraction, if necessary, some recess could be incorporated in the wet deck bottom.

(3) The proposed multi-lobe rear trunk was considered as the simplest responsive arrangement and one which could be retracted most easily. Its behaviour in negotiating broken ice fields would need to be examined in detail as the design is developed further.

Yours sincerely,
BRITISH HOVERCRAFT CORPORATION LTD.

T.F. Arlotte

T.F. ARLOTTE
Chief Dynamics Engineer

ANNEX B

Telex from the British Hovercraft Corporation Limited
to German & Milne Inc.

P4339, dated 28 October 1981

TO: GERMAN & MILNE INC., MONTREAL, CANADA
FROM: BRITISH HOVERCRAFT, COWES, IOW, ENGLAND
REF: P4339

29.10.81

COPY: T.K.S. MURTHY (TO DYNAMICS DEPT.) BHC

FOR ATTENTION OF: J. GORDON GERMAN

AFTER FURTHER DISCUSSION OF OUR REPORT IFA/ANK/PC/2769 DATED 8 SEPT. WITH DR. MURTHY, WE FEEL THAT A TOTAL POWER OF 62800 HP SHOULD BE INSTALLED FOR THE AIR CUSHION SYSTEM, ALTHOUGH THE POWER ABSORBED COULD BE MINIMISED BY REDUCING LIFT FAN RPM UNDER FAVOURABLE CONDITIONS. SEPARATE LIFT MACHINERY ON THE DECK IN THE FORM OF MARINISED GAS TURBINES IS AN ALTERNATIVE. BEHAVIOUR OF SKIRT FINGERS OVER ICE HAS BEEN FOUND SATISFACTORY IN SRN6 COLD WEATHER TRIALS IN NORWAY AND SWEDEN AT -14 DEGREES C AND OTHER TRIALS OVER ICE.

MACHINERY FOR RETRACTING SKIRT COULD BE ELECTRIC MOTORS DRIVING CAPSTAN TYPE ROLLERS ON COMMON SHAFT THROUGH WORM GEAR.

CHANGE IN LOADING ON FAN BLADE DURING PITCHING MAY CONSTITUTE NO PROBLEM DUE TO INERTIA OF THE SYSTEM RESTRICTING CHANGE IN FAN SPEED WITH FLOW CONDITIONS.

DR. MURTHY LOOKS FORWARD TO RECEIVING YOUR PRELIMINARY DRAFT REPORT FOR HIS ACTION.

REGARDS

T.F. ARLOTTE
PP TKS MURTHY

ANNEX C

MINORITY REPORT BY DR TKS MURTHY

SSACV ICEBREAKING LNG TANKER

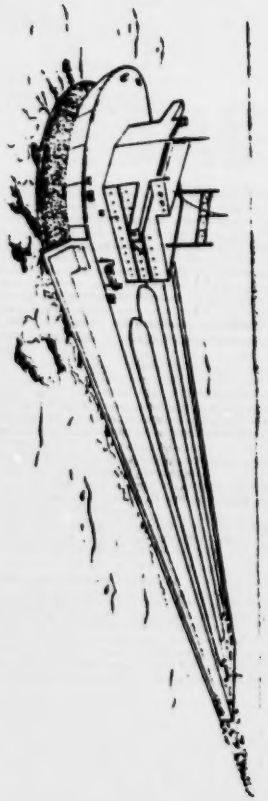
Notes on Design and Predicted Performance

by

T K S Murthy

Naval Architect & SSACV Development Consultant

December, 1981



SSACV Icebreaking LNG/OIL TANKER

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B. Computer Plot of Air Cushion Wave Resistance from the British Hovercraft Corporation

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IV Minimum Radius of Lead and Distance of Circumferential Cracks (after Carter) 7
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Figure 1 Installed Power Required for Various Speeds in Calm Water (Loaded and Empty Conditions)
2 Total Installed Power Required (Lift and Propulsion) for Various Ice Transiting Speeds (Loaded and Empty Conditions) at Different Cushion Pressures

SSACV ICEBREAKING [illegible] TANKER

1. Design Philosophy and Selection of General Dimensions

For the purpose of this preliminary feasibility study, the leading particulars indicated in Table I have been tentatively selected on the basis of the design considerations described below. This geometry may not necessarily provide the optimum configuration for efficient ice transiting and economical movement in open water, but suitable modifications may easily be carried out during the course of further theoretical study to be ultimately confirmed or corrected by scale model tests.

1.1. Upper Hull

The upper hull will be rectangular in planform with a roughly semi-circular bow suitably shaped for ridged-ice cutting, perhaps on the lines of the spoon-shaped bow adopted by Dome Petroleum for their Arctic 3 icebreaking offshore supply ship (The Motor Ship, October 1981), and located at a suitable height above the ice level for this purpose. This height may also be adequate in open water for the prevention of the impingement of green water on its lower surface, the so-called "wet deck". The hull with its raft structure will have sufficient depth so that it will act as a buoyancy chamber with adequate reserve buoyancy in the case of a failure of the air cushion system combined with the unlikely case of a simultaneous flooding of the side struts and the submerged hulls. Thus, the vessel will now be able to float safely over water. In ice covered regions, the wet deck will be of sufficient strength to enable the vessel to land and rest on the ice layer without any damage to the structure. The ice will naturally be fractured by the application of uniform pressure of considerable magnitude over a large area, but this type of icebreaking is, of course, not recommended!

1.2. Side Struts

The side struts connecting the submerged hulls to the upper structure are assumed to be parallel-sided over the middle half of their length and with lenticular ends (in planform section) of equal length. The thickness of the struts is assumed to vary linearly from a minimum at the lower end carrying the submerged hull to a maximum at the upper end in order to provide adequate space for suitable structural members at the root joint of the cantilever-type buoyancy members. The vertical slope of of the sides of the struts will also enable the normal pressure of the ice on the plating to be avoided. Another reason for this type of tapering is to increase the stiffness of the vessel in heave over water as measured in tons/inch or kg/cm.

The stems of the struts will be ice-reinforced and suitably designed for ice cutting. It is suggested that a vee-shape may be intoduced in the cutting edge with apex at the iceline, i.e. on the surface of the ice sheet, in the hope that the ice will be cut downwards in bending and again upwards when pitching oscillations of the vessel are induced by a suitable deployment of the flexible seals containing the air cushion. It is realised that the corner of the vee may actually be choked with ice, in which case we will have to rely on crushing the ice layer, although the bending action can still be carried out by the upper and lower arms of the vee free from ice congestion.

The struts will be as narrow as possible to reduce ice resistance, but with sufficient waterplane area to provide adequate transverse stability.

1.3. Submerged Hulls

The submerged hulls are assumed to have the shape of a circular cylinder over 90% of their length with a hemi-spherical nose end and a conical stern.

This is perhaps not the best hydrodynamic shape for laminar flow conditions and, in any case, the pointed stern would not provide adequate room for the machinery and transmission. At a later stage we may have to examine the possibility of altering this to a truncated cone of, say, elliptic cross-section in order to provide a transom stern suitable for the installation of a pair of possibly contra-rotating propellers in each hull, particularly for the higher power levels we will be considering (100 and 150 MW).

The choice of a suitable diameter, D_h, for the hulls is based on the following considerations:

(i) D_h should be sufficiently large so that the total displacement of 140,000 t in the fully loaded condition would not call for an excessive air cushion lift in the ice regions or an excessive draught in open water;

(ii) D_h should not be very large as there should be some scope for the use of an adequate amount of cushion pressure (of the order of 400 psf) for icebreaking purposes;

(iii) D_h should be of sufficient magnitude that when the strut thickness is also taken into consideration, it will be possible to vary the draught arbitrarily within some wide limits by a reduction of the cushion pressure from its standard value and a readjustment of the depth of the flexible seals below the hard structure; etc.

1.4 Air Cushion System

The air cushion bounded laterally by the side struts will substantially be of rectangular planform. However, the bow seal will roughly be of semi-circular shape with a rounded upper bag carrying a multitude of fingers and following the contour of the overhanging upper hull with its own ridged-ice cutting bow. Thus the bow seal will be extending well forward of the stems of the side struts except at the actual points of attachment thereon. With this substantial extension over the main part of the air cushion we may hope that the usual radial and circumferential cracks will appear in the ice layer just ahead of the points where the cutting edges of the side struts enter the ice, thus facilitating ice breaking and ice transiting. This geometry for the bow seal may call for tapered fingers close to the points of attachment to the structure and some special attention will also have to be given to the mechanism of skirt retraction.

The rear skirt will be located at the stern of the side struts so that the centre of pressure may lie below the centre of gravity of the vessel. Any forward location of the rear seal will shift the centre of pressure ahead of the centre of gravity and although this could be catered for by a suitable weight distribution, difficulties will arise when the vessel is operated in open water with the air cushion inoperative.

For the sake of simplicity we will assume that the side struts and the submerged hulls are of the same length with the mean length of the air cushion being slightly higher, although it may be considered prudent to have the struts overhanging the hulls at the front in order to minimise damage to the latter from submerged ridges. A similar overhang may also be desirable at the rear so that damage to the propellers may be avoided when it is necessary to reverse the vessel, say, for a charging operation.

The cushion area is substantially determined by the overall dimensions and the cushion pressure also specified from the thickness of the ice layer to be broken by the application of this pressure, preferably with an air cavity produced under the ice layer. The air cushion lift is therefore determined

and it will be necessary to manipulate the dimensions of the buoyancy members such that suitable draught conditions may be achieved by an adjustment of the cushion pressure under various operational conditions, such as laden and empty conditions, level and ridged ice with ridge sails elevated above and the ridge keels submerged below the general ice level, open water operation, etc.

Any reduction of pressure below the standard value (405 psf for a 7 ft ice layer) will not produce a cavity under the ice layer and it will also not be possible to bleed air from the cushion for the purpose of lubricating the outer surfaces of the side struts against ice friction where they penetrate the ice. However, some icebreaking will be possible even with lower pressures as will be discussed below.

The bow and stern seals of the air cushion will be designed in such a manner that they can easily be raised or lowered to lie in close proximity with the ice surface following the variation of cushion pressure and they will also have the capability of being completely retracted and stowed in a recess in the upper hull for open water operation. It may be useful for this purpose to employ the scheme of overlapping elements in the skirt fingers as disclosed in U K Patent Specification No.1,574,800 assigned to Dr T K S Murthy.

Table I shows the general particulars of the SSACV Tanker. The dimensions have been tentatively selected as a compromise between the various conflicting design requirements discussed above without any serious attempt at optimisation at this stage.

Table II gives the derived values of buoyancy, wetted surface and air cushion lift at various strut immersions

Table III gives similar values for some selected icebreaking and open water configurations in the laden and empty conditions. These configurations have not yet been checked for compliance with stability requirements. The values for cushion volume flow and lift power have been supplied by the British Hovercraft Corporation for the design cushion pressure of 405 psf assuming an equivalent air gap of 4.5 inches and allowing an additional 5% for lubrication purposes. These values have strictly been scaled down in the ratio of $p_c^{1/2}$ for volume flow and $p_c^{3/2}$ for lift power at reduced pressure values, but may have to be increased slightly for the rounded extension at the bow.

Table III also gives the shaft power available for propulsion on the basis of total installed shaft power levels of 70,100 and 150 MW. It is of course assumed that lift and propulsion power are integrated in the machinery.

2. Icebreaking Performance

2.1 Air Cushion Icebreaking

The air cushion contained substantially by the side struts in the lateral direction will have a rounded bow extending well ahead of the struts where they enter the ice layer. The width of the cushion will vary from 51 to 53m depending upon the depth of immersion of the struts (see Table III) and the semi-circular bow portion will extend some 26m beyond a line connecting the stems of the struts.

The mean cushion length is taken to be 330m and as the Froude number, F_ℓ, based on length is only about 0.1 even at 10 or 12 knots over ice, we will only be concerned with low speed or almost quasi-static icebreaking.

We will be using a standard cushion pressure of 405 psf (2.81 psi/19.39 kPa) to balance the buoyancy forces supporting the 7 ft thick ice sheet. This will enable an air cavity to be produced under the layer for efficient icebreaking. Also, it will be possible to bleed air from the cushion and

TABLE I

General Particulars of SSACT Tanker

Δ_{tot}	Displacement, laden (t)	140,000	Eastward voyage to unloading ports
	empty (t)	82,810	Westward to loading terminals
	Cargo capacity (cu m)	140,000	Four 13m dia. x 250m long containers in upper hull
Δ_{130}	Cargo deadweight, laden (t)	60,200	5% retention in cryogenic tanks
	empty (t)	3,010	
	Upper Hull		
L_{uv}	Length (m)	350	Substantially rectangular in planform and with a rounded bow suitable for ridged-ice cutting
B_{uv}	Beam (m)	73.4	Adequate reserve buoyancy
H_{s}	Depth (m)	6.0	
	Side Struts		
L_s	Length (m)	320	Parallel middle portion with lenticular ends
b_1	Width, top (m)	10.0	
b_2	bottom (m)	6.0	
b_w	waterline (m)	6 -10	
b_i	iceline (m)	6.43-10	Width of strut on the ice surface
b_s	Centre-line separation (m)	59.4	
d_s	Depth of immersion (m)	0 -20	
$d_{s(max)}$	Total depth (m)	20	Depth from "wet deck" to submerged hull attachment
C_w	Waterplane area coefficient	0.83	
	Submerged Hulls		
L_h	Length (m)	320	Mostly cylindrical with hemispherical nose and end conical tail piece
D_h	Diameter (m)	14.0	
	Centre-line separation (m)	59.4	
	Centre-line submergence (m)	various	
D_h/L_h	Diameter/length ratio	0.04	
	Separation/diameter ratio	4.20	
C_p	Prismatic coefficient	0.93	
	Air Cushion		
L_c	Length, mean (m)	330	Rectangular in planform and with a rounded bow [illegible] breaking ice ahead of the side struts
B_c	Beam (m)	51-53	
H_c	Height (m)	8.3-20	
	Aspect ratio	0.16	
S_c	Cushion area (sq m)	16962	Average value for strut immersion of 10 m
p_c	Cushion pressure (psf)	405	Standard pressure can be varied to lower values by reducing the speed of the lift fans and retracting the skirts accordingly
	(standard) (psi)	2.81	
	(kPa)	19.39	

lead it to the outer surfaces of the side struts below the ice layer for lubrication against ice friction. The inner surfaces will obviously be similarly lubricated directly by the air cushion through cracks in the ice.

If we use a pressure below the minimum of 405 psf required for icebreaking with a cavity for the sake of conserving lift power, it will still be possible to break ice against the resistive forces due to buoyancy when there is no cavity, but the radius of load will have to be much larger. Similarly, the predicted distance for the appearance of circumferential cracks in the ice sheet will also be greater. But these are adequately catered for by the large dimensions of our air cushion (330m x 52m).

TABLE II

Buoyancy, Wetted Surface and Air Cushion Lift

d_s	Side strut immersion (m)	2	4	6	8	10	12
H_c	Height of air cushion (m)	18	16	14	12	10	8
b_w	Waterline width of strut (m)	6.4	6.8	7.2	7.6	8.0	8.4
	Mean immersion of strut (m)	6.2	6.4	6.6	6.8	7.0	7.2
B_c	Width of air cushion (m)	53.0	52.6	52.2	51.8	51.4	51.0
s	Buoyancy, both struts (t)	6779	13995	21648	29739	38627	47232
h	Buoyancy, both hulls (t)	94143	94143	94143	94143	94143	94143
b	Buoyancy, total (t)	100922	108138	115791	123882	132770	141375
c	**Air cushion lift** (t)						
	Cushion pressure-405 psf	34580	34327	34066	33805	33544	33283
	300	25621	25427	25234	25041	24847	24654
	200	17081	16952	16823	16694	16565	16436
	100	8540	8476	8411	8347	8282	8218
	0	0	0	0	0	0	0
tot	**Total displacement (t)**						
	Cushion pressure-405 psf	135510	142465	149857	157687	166214	174658
	300	126543	133565	141025	148923	157617	166029
	200	118003	125090	132614	140576	149335	157811
	100	109462	116614	124202	132229	141052	149593
	0	100922	108138	115791	123882	132770	141375
d_s/D_h	Strut immersion/hull diameter ratio	0.14	0.29	0.43	0.57	0.71	0.86
T	Draught (m)	15.3	17.3	19.3	21.3	23.3	25.3
$S_{w(s)}$	Wetted surface, struts (m^2)	2795	5355	7915	10475	13035	15595
$S_{w(h)}$	Wetted surface, hulls (m^2)	23842	23842	23842	23842	23842	23842
$S_{w(tot)}$	Wetted surface, total (m^2)	26637	29197	31757	34317	36877	39437

Some calculations have been made using the theoretical results derived by Carter(1) for static icebreaking with and without an air cavity under the ice layer. The results are shown in Table IV.

These theoretical results have been used with some reservation in view of the fact that the ice failure model comprising a series of semi-infinite wedges adopted by Carter was not experimentally confirmed in the ACT-100 full scale icebreaking trials conducted in 1975/76 and reported by Transport Canada(2) in TP 1984(page 36). It appears that in practice the ice tends to fail as a single beam across the width of the air cushion and then break into pieces as it fails. However, in the absence of any other suitable model validated by experiment, we may continue the discussion following Carter's study.

Firstly, it appears in our case that even a cushion pressure as low as 200 psf could break 7 ft of ice as the cushion is of very large dimensions and the load is applied over a very large area. The minimum radius of load is 23.1 m in this case and the semi-circular part of the cushion at the rounded bow with a radius of 26 m could by itself be adequate for this purpose, with the circumferential cracks appearing, possibly, 37.2 m ahead of the base of the semi-circle, i.e. the line joining the cutting edges of the side struts on the ice layer. However, our cushion will be very long, with a mean length of 330 m and we may expect a series of radial and circumferential cracks to be propagated within the air cushion extending to the side struts which are spaced 52 m apart. This may be seen from an examination of the entry of the SSACT from open water to a semi-infinite ice layer. When the bow portion of the air cushion surmounts the ice, the stems of the side struts will be 3 m

TABLE III

Selected Icebreaking and Open Water Configurations (Laden and Empty Conditions)

	ICEBREAKING								OPEN WATER		
	Laden	Empty	Laden	Empty	Laden	Empty	Laden	Empty	Laden	Empty(1)	Empty(2)
Cushion pressure (psf)	405	405	300	300	200	200	100	100	0	0	0
Cushion air flow (cu m/s)	58000	58000	49918	49918	40758	40758	28820	28820	0	0	0
Lift power (hp)	62800	62800	40037	40037	21793	21793	7705	7705	0	0	0
Strut immersion (m)	3.5	3.5	6.0	4.0	8.0	4.0	10.0	4.0	11.7	4.0	8.0
Height of cushion (m)	16.5	16.5	14.0	16.0	12.0	16.0	10.0	16.0	8.3	16.0	12.0
Strut width waterline (m)	6.7	6.7	7.2	6.8	7.6	6.8	8.0	6.8	6.3	6.8	7.6
iceline (m)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	-	-	-
Draught (m)	16.8	16.8	19.3	17.3	21.3	17.7	23.3	17.7	25.0	17.3	21.3
Width of cushion (m)	52.7	52.7	52.2	52.6	51.8	52.6	51.4	52.6	-	-	-
Buoyancy, struts (t)	12150	12150	21648	13995	29739	13995	38627	13995	45860	13995	29739
Buoyancy, hulls (t)	94143	94143	94143	94143	94143	94143	94143	94143	94143	94143	94143
Buoyancy, total (t)	106293	106293	115791	108138	123882	108138	132770	108138	140003	108138	123882
Air cushion lift (t)	34392	34392	25234	25427	16694	16952	8282	8476	0	0	0
Water ballast (t)	(685)	(57875)	(1025)	(50755)	(576)	(42280)	(1052)	(33804)	0	(25328)	(41072)
Net displacement (t)	140000	82810	140000	82810	140000	82810	140000	82810	140003	82810	82810
Wetted surface, struts (sq m)	4715	4715	7915	5355	10475	5355	13035	5355	15211	5355	10415
hulls (sq m)	23842	23842	23842	23842	23842	23842	23842	23842	23842	23842	23842
total (sq m)	28557	28557	31757	29197	34317	29197	36877	29197	39053	29197	34317
Propulsion power (hp) available with an installed power of											
70 mW	31072	31072	53835	53835	72079	72079	86167	86167	93872	93872	93872
100 mW	71302	71302	94065	94065	112309	112309	126397	126397	134102	134102	134102
150 mW	138353	138353	161117	161117	179360	179360	193448	193448	201153	201153	201153

NOTE These configurations have not been checked for compliance with stability requirements.

behind in open water. We may assume that the circumferential cracks will appear 14 m ahead of the bow seal of the air cushion. The side struts will therefore have to cut their own path in the solid ice over the first 40 m or so, although during this passage the main part of the cushion will be gradually mounting the ice and producing similar cracks some 37 m ahead of the rear edge of that part of the cushion directly over the ice layer. This means in effect that the side struts will virtually be entering broken ice.

TABLE IV

Minimum Radius of Load and Distance of Circumferential Cracks in 7 ft thick ice layer for various cushion pressures

Cushion pressure, P_c	(psf)	405	300	200	100
P_c	(psi)	2.81	2.08	1.39	0.69
Minimum length of air cushion	(m)	13.1	15.7	23.1	No failure
Distance of circumferential cracks from rear of cushion	(m)	27.3	29.9	37.2	No failure

Reverting to icebreaking with a low cushion pressure of 200 psf and a 23.1 m minimum . of load, it may be argued that as the forward part of the air cushion is mainly responsible for icebreaking, a cushion only about 25 m long would be adequate and economical for the purpose. But the volume flow of air and lift fan power are based entirely on the escape area at the front and rear for a given cushion pressure irrespective of the length of the cushion. Also, if we consider a 330 m long air cushion as a connected series of overlapping cushions each 25 m long, when the first part moves over the ice without breaking it, the following parts may have an opportunity of doing so. The breaking of the ice inside the full length of the cushion would therefore greatly reduce the ice friction of the side struts and would improve their passage through the ice layer.

It will also be noted that if a cushion is provided at the forward part of the vessel only, there will be a destabilising moment, particularly when the cushion pressure is varied. This could be compensated for by a suitable load distribution in the vessel. But in the case of open water we will not be employing the air cushion at all and difficulties will arise.

It is therefore obvious that the air cushion should extend substantially over the length of the vessel with its centre of pressure, the centre of buoyancy of the submerged members and the centre of gravity of the entire vessel all lying along the same vertical line.

There are also other considerations. It would be easier to attach and maintain the flexible seals of the air cushion at the front and rear of the vessel. Also, we are exploring the possibility of oscillating the SSACV Tanker in pitch and bodily raising or lowering the vessel in heave by a suitable adjustment of the seals. We are particularly interested in raising the superstructure above elevated ridges and lowering the hulls deeper below submerged ridges. All these operations will require an air cushion lift of substantial magnitude obtainable by using a large cushion area and a high cushion pressure, so that by controlling (or cancelling) the air cushion lift by adjusting the lift fan rpm, we may be able to vary the draught of the vessel within fairly large limits.

A high cushion pressure will also be necessary if we consider the lubrication of the outer surfaces of the side struts against ice friction. There is also the possibility of venting cushion air through "puff ports" to produce side forces for manoeuvrability in ice, so that once again a large volume flow of air will be required.

We may therefore summarise the above discussion in the following manner:

(i) The air cushion will extend substantially between the side struts and with an extension in the form of a rounded bow section;

(ii) The standard cushion pressure of 405 psf(2.81 psi) will enable efficient icebreaking of a 7 ft ice layer to be carried out. It will also be possible to bleed air through the side struts to the outer surfaces for lubrication against ice friction. The inner surfaces will similarly be lubricated directly by the air cushion;

(iii) Icebreaking will also be possible with a reduced pressure of 300 or even 200 psf in view of the very large cushion area, but it will not now be possible to effect lubrication of the outer strut surfaces in the ice layer. The inner surfaces will be partially lubricated directly by the air cushion. However, the use of a low cushion pressure will enable more power to be released for propulsion (see Table III);

(iv) Although it is not strictly essential for the ice-transiting tanker to break ice between the struts compared with an icebreaker, the use of a cushion pressure even of moderate value for icebreaking will enable the vessel to move with reduced friction on the side struts immersed in the ice layer. The breaking of ice on the inboard side between the struts will also assist manoeuvrability in ice; and

(v) A large value of the cushion pressure implies a large momentum flow of the air escaping from the cushion, particularly under the bow seal. This momentum of the air will be useful in submerging the broken slabs of ice under the unbroken ice layer. Similarly, the momentum flow under the rear seal will clear the stern region by pushing the slabs downstream out of the way of the propellers at the rear of the submerged hulls. These forces causing submergence or clearance of the ice pieces are produced by the air cushion and <u>are not reacted by the vessel</u> unlike the case of the conventional icebreaker where the forces are reacted by the hull. It may therefore be expected that ice resistance will be much less in our case as discussed in the section which follows.

2.2 Side Strut Icebreaking (Continuous Mode)

The side struts have an iceline width (i.e. width at the top of the ice layer) varying from 7.13 to 8.43 m depending upon the depth of their total immersion in ice and water. Taking an average value of 8.0 m the thickness to length ratio is only 0.025 and the prediction of the icebreaking ability of such thin struts pushing into and breaking solid ice with a large amount of propulsive power has to be carried out with caution in view of possible buckling instability.

We will use two different formulations of ice resistance in continuous mode icebreaking of level ice derived by Lewis and Edwards(3) and Edwards *et al*(4) respectively:

$$R_i = 0.146\,\sigma_f h^2 + 8.840\,\rho_i g B h^2 + 5.905\,\rho_i B h V^2 \qquad \text{..............(1)}$$

$$R_i = \rho_i g B h^2 \left[4.242 + 0.05033\,\frac{\sigma_f}{\rho_i g h} + 8.948\,\frac{V}{\sqrt{gh}} \right] \qquad \text{..............(2)}$$

The first semi-empirical equation is based on a regression analysis of model and full-scale ice resistance data. The first term on the R.H.S. accounts for all forces of resistance involved in the breaking of the ice layer into slabs, i.e. the mechanical forces of reaction and frictional forces. The second term accounts for all resistive forces due to the buoyancy of the ice slabs, namely, the work required to tip and submerge the broken ice pieces under the unbroken ice field and the frictional forces caused by the broken ice passing

the side struts. The last term represents the resistive forces due to the momentum exchange between the vessel and the broken ice in imparting a velocity to the broken pieces as the vessel moves through the ice cover.

It appears that the resistive forces associated with the breaking the ice is only a small part of the total (Reference 3, page 231) and that a particular value selected for the flexural strength, σ_f, of the ice layer may not substantially affect the predicted total ice resistance. The last term representing momentum exchange seems to be of some more importance, but it is the second term which has a dominant effect. This term (like the first) includes frictional forces between the struts and the ice. It is also confirmed in Reference 4 (pages 374,406) that the strength of the ice has little effect and that the friction factor has substantial effect on ice resistance. On the other hand, it is reported in Reference 2 that hard high tensile strength ice fails much more readily than weak soft ice of the same thickness.

The second equation has been selected as simpler than a more complicated equation derived by the same authors involving some other vessel parameters on page 377 of Reference 4. This equation has also been derived by regression analysis, but the first term proportional to Bh^2 is of the same form as the second term of the first equation, although the coefficient has only half the value. The second term deals with icebreaking forces and the last term is due to the inertia of the broken ice pieces.

We may carry out ice resistance calculations with the separate application of the two predictors quoted above, but we may modify some of the coefficients in our favour due to the icebreaking capability of the air cushion between the struts discussed in Section 2.1 above in the following manner:

Terms relating to			Modification of Coefficients	Remarks
Icebreaking	Eqn 1, first term		0.146 replaced by 0.073	If side struts enter [illegible] will be less. Frictional forces will also be less if the surface of the surface of the ice layer is lubricated by cushion air during ice cutting. When [illegible], no air cushion icebreaking will be possible, but some alleviation of strut-ice friction is possible on the ice surface.
	2, second term		0.05033 " " 0.025	
Submergence of ice slabs	Eqn 1, second term		8.840 replaced by	Lubrication of inboard surfaces of side struts (possible for all P_c) and outboard surfaces in the case of P_c = 405 only greatly reduces strut-ice friction.
	P_c	405	3.5	
		300	4.5	
		200	5.5	
		100	6.5	
	Eqn 2, first term		4.242 replaced by	Momentum flow of air at the bow seal will assist in submerging the ice slabs under the ice layer.
	P_c	405	1.7	
		300	2.2	
		200	2.6	
		100	3.5	
Momentum exchange	Eqn 1, third term		5.905 replaced by	Momentum flow of cushion air will impart the necessary velocity to the broken ice pieces to clear them away from the side struts. This force is not reacted by the vessel unlike conventional icebreakers.
	P_c	405	3.0	
		300	4.0	
		200	5.0	
		100	5.5	
	Eqn 2, third term		8.948 replaced by	
	P_c	405	4.5	
		300	6.1	
		200	7.6	
		100	8.0	

The coefficients have been arbitrarily reduced above on a sliding scale depending upon the cushion pressure used. This can only be confirmed by model tests.

Whilst the empirical and analytical predictors will be retained in the form of equations (1) and (2) above when the air cushion is inoperative, i.e. when p_c is zero, we may use the equations with the coefficients modified on a sliding scale for the respective cases of p_c=405, 300, 200 and 100 psf.

The calculations assume a standard icebreaking shape for the side struts moving horizontally and cutting ice downwards in the conventional manner. However, if we accept that there is a capability for the SSACV Icebreaker to be forcibly oscillated in pitch by a manipulation of the flexible seals of the air cushion and if the cutting edges of the struts have a vee-notch at the ice level, the efficiency of icebreaking may be substantially improved, as it will now be possible to ice downwards in the conventional manner and again upwards, which is a new capability. This device may be particularly useful in the cases of weak soft ice, i.e. of low tensile strength, where it is believed that the ice simply bends without breaking.

2.3 Upper Hull Ridged-Ice Cutting (Qualitative Discussion)

Although we are only concerned with continuous breaking of 7-ft thick level ice in this study, it may be hypothetically recorded in qualitative terms for future reference that it may be possible for the SSACV Icebreaker to cut and move through ridged-ice because of its special configuration.

We have proposed (see Table III) that in the fully loaded condition the SSACV will have a draught of 16.8 m with the side struts immersed to a total depth of 5.63 m(3.5 m in water and 2.13 m in ice). The upper hull will therefore be 14.37 m above the ice level and we may expect the vessel to clear ridged-ice with a sail height approaching, say, 12 or 13 m above the general level. The rounded bow section will be of a shape suitable for icebreaking and ridged-ice cutting will be possible by a small downward inclination of the vessel in pitch. In the case of very strong pressure ridges, it may be possible to apply a large downward force at the bow by reducing the cushion pressure drastically or even collapsing the cushion altogether, thus making the vessel sink in heave.

Similarly, it will be possible for the submerged hulls to clear ridges of moderate keel depth extending below the general level. If the lift fans are stopped, the draught will increase from 16.8 to 25.0 m, so that submerged ridges some 7 or 8 m thick can be negotiated.

However, these are only hypotheses at the moment and detailed calculations and model tests will be necessary. A confirmation of this capability of the SSACV Icebreaker in ridged-ice conditions will be invaluable as conventional tankers require the most icebreaking assistance in this type of ice environment.

3. Performance Calculations

3.1 Machinery and Propulsion

It is assumed that machinery for air cushion lift and for propulsion will be integrated, so that when the air cushion is used at reduced pressures or not at all as in open water, additional power will be released for propulsion as indicated at the bottom of Table III. Gas turbine machinery may be an obvious choice as they could use the "boil off" from the cryogenic tanks as fuel. Electric transmission, in spite of its poor efficiency, may be the answer to the problem of transmitting a large amount of power by mechanical means to the lift fans in the upper hull from the machinery in the submerged hulls. Shock loads may also be absorbed easily by the electric motors.

Propulsion will be by conventional CP water screws and a compromise in their design has to be made between the need for a high bollard thrust during ice cutting at speeds increasing slowly from zero and a reasonably high efficiency at the free running design speed in open water. Nozzles will naturally

be provided for the protection of the propellers in the ice regions and also for a high bollard pull and larger thrust at low forward speeds, although this implies that propulsive efficiency will be sacrificed at the free running design speed in open water.

The assumption of the magnitude of the available propulsive coefficient is always a bone of contention in the absence of controlled model tests. However, Messrs Stone Vickers Limited have provided at the author's request some estimated propeller particulars (Annex A). On the basis of this information and on the assumption of a cubic variation of the propulsive coefficient, η, defined as the EHP/SHP ratio, the form of the cubic equation has been determined

$$\eta = a_1 V_m\left(\frac{V}{V_m}\right) + (3\eta_m - 2a_1 V_m)\left(\frac{V}{V_m}\right)^2 + (a_1 V_m - 2\eta_m)\left(\frac{V}{V_m}\right)^3 \quad \text{.......(3)}$$

where η_m is the maximum value of at the design speed V_m (in feet per second) and a_1 is related to the bollard thrust in the form

$$a_1 = \frac{\text{Bollard thrust(lbs)}}{550 \times \text{SHP(British)}}$$

This equation enables the estimation of the propulsive coefficient at various free running speeds in open water and at low ice-transiting speeds. The values obtained by this method are slightly higher than those estimated by Stone Vickers for the ice region (0.30 > 0.251 at 4.8 knots with twin screws and 0.337 > 0.292 at 6.9 knots with quadruple screws). In any case, these data are probably strictly true only for the two propeller designs for the two levels of SHP, but we may assume that they are applicable even for other power levels. The assumed values of the propulsive coefficient are plotted in Figures 1 and 2.

3.2 Performance in Open Water (Calm Conditions)

It will be seen from Table III that we are proposing a strut immersion of 11.7 m (draught 25.0 m) in the loaded condition and two lower values 4.0 m(draught 17.3 m) and 8.0 m(draught 21.3 m) in the empty condition with a ballast of 25328 and 41072 t respectively. The air cushion will not be used in open water operation and the flexible seals will be fully retracted.

The prediction of power requirements in calm water is straightforward. The skin friction coefficient, C_f, of the buoyancy members is calculated from the ITTC(1957) formula with an additional roughness allowance of 0.0004. The residuary resistance coefficient, C_r, is obtained by interpolation from the computed results provided by CETENA (Reference 5) in Italy for various strut immersion/hull diameter ratios applicable to NSRDC Model No.5287, which is roughly of similar proportions. The C_r values are very low indeed, smaller than the roughness allowance of 0.0004 we have included, even at a Froude number of 0.23(25 knots). There are peaks and troughs in the C_r values at these very low speeds, but the mean values have been used in the calculations for open water conditions and ignored subsequently as trivial in the case of ice regions.

Figure 1 shows the installed power requirements of the SSACV Tanker in the loaded and empty conditions at various speeds in calm water with the inclusion of the assumed values of the propulsive coefficient (EHP/SHP).

3.3 Rough Water Operation

It is customary to use calm water performance as a baseline for the comparison of the merits of different hull forms of similar type. This type of comparison is perhaps valid for conventional type of displacement vessels, but our configuration with submerged buoyancy members carried by surface-piercing side struts of small waterplane area is radically different and is expected to be less subject to wave excitation in rough seas. The motions

of the vessel are therefore expected to be less, giving it the ability to maintain its design speed in rough water conditions. It is understood from Reference 6 that in this simple SWATH configuration the SSACV Tanker will require a relatively small power increase of the order of 2 to 8% in head seas up to Sea State 5 ($H_{1/3}$=10 ft/3.1m). This is also confirmed by Kamen (Reference 7, Figs 6 and 7), where the speed decrement of a 30-knot vessel is only about 2 knots up to Sea State 6.

It is understood that the SSACV Tanker will be operating in open water where North Atlantic sea conditions prevail, i.e. Sea State 4 ($H_{1/3}$=5.5 ft) and higher sea states occurring 70% of the time (Reference 6).

We may therefore assume that the power required to maintain speed in, say, Sea State 5 will be increased only by about 10% compared with a possible increase of the order of 40-50% for conventional displacement vessels.

3.4 Performance in Ice Regions

The air cushion will be used in ice regions and we have proposed icebreaking operation at the standard cushion pressure of 405 psf and at pressures well below this value.

The resistance in ice regions is composed of:

- (i) Resistance due to the air cushion moving over the ice layer;
- (ii) Resistance of the flexible seals possibly contacting the ice cover;
- (iii) Resistance of the buoyancy members moving through water under the ice layer; and
- (iv) Resistance of the side struts moving through the ice layer, breaking the ice and dispersing the broken pieces.

As regards (i) above, the components are the momentum drag and the wave resistance. The former has been calculated for one of the configurations and found to have a negligible value even though the cushion pressure was the highest (405 psf) and the speed was taken as 12 knots. The wave resistance has also been calculated with imposed wave steepness limitations according to the formula applicable to very low forward speeds

$$R_{w(max)} = \rho_w M_S^2 B\, V_a^4 / 196 \quad \text{..............(4)}$$

and found (as was expected) to have extremely low values up to 12 knots. This is also confirmed by a computer-plotted figure supplied by the British Hovercraft Corporation (Annex B) for the case of

$$L_c = 800 \text{ ft}; \quad B_c = 69 \text{ ft}; \text{ and } \quad p_c = 355 \text{ psf}$$

and it was therefore decided to ignore the air cushion wave resistance in further calculations.

As regards (ii) above, it is not possible to estimate skirt drag from theory. Model tests will be necessary.

the calculations relating to (iii) and (iv).
are plotted in Fig.2 showing the level of total installed power required for lift and propulsion at various ice-transiting speeds in the continuous mode of operation over ice regions 7 ft(2.13 m) thick. The flexural strength of ice is taken as 150 psi.

Fig. 2 also shows the assumed values of the propulsive coefficient as discussed in Section 3.1 above.

4. Summary of Results and Conclusions

The design philosophy of the SSACV Tanker and the selection of its general dimensions are discussed in Section 1. The leading particulars of the vessel are given in Table III. In open water, the air cushion will not be used and the vessel will have a loaded draught of 25.0 m. The draught will be reduced to 17.3 m with a ballast of 25320 t or 21.3 m with a heavier ballast of 41072 t. In ice regions, the vessel will have a draught of 16.0 m both in the fully loaded and empty conditions when the standard cushion pressure of 405 psf is used for icebreaking. This draught will be increased when reduced pressures are used.

The icebreaking performance of the vessel with various values of cushion pressure is discussed in Section 2.1. It has been demonstrated that even moderate values of the pressure will be effective in breaking the ice as the cushion area is very large indeed.

The ice resistance of the side struts in continuous mode icebreaking is discussed in Section 2.2. Two different formulations of the ice resistance obtained by regression analysis have been selected from the literature and both used for the calculations giving remarkably close results. Also, the coefficients of terms in the equations related to different aspects of the icebreaking and ice dispersal process have been modified on an empirical basis to take into account the icebreaking capability of the air cushion in this respect, perhaps of varying effectiveness, depending upon the value of the cushion pressure used.

The ridged-ice breaking capability of the upper hull is briefly discussed in Section 2.3.

Fig.1 illustrates the level of installed power required at various speeds in open water under calm conditions both for the fully loaded and empty configurations. As discussed in Section 3.3, these values may be increased by 10% for rough water operation, say, up to Sea State 5.

Fig.2 illustrates similar level of total installed power (lift and propulsion) at various ice transiting speeds in 7-ft level ice for the continuous mode of progression with varying amounts of cushion pressure both for the laden and empty conditions.

Table V summarises the significant final results, namely, the maximum speed in open water and the maximum ice transiting speed, for three different levels of total installed power (70, 100 and 150 mW).

It is clear from these results that with a low power installation of 70 mW intended for low ice transiting speeds (say, 4-6 knots) the use of a low cushion pressure of 200 or even 100 psf will give us a speed of 6.2 knots, some 2 knots higher than that achievable without the air cushion. This is in the loaded condition. When the vessel is empty, the use of a low cushion pressure (100 psf) again gives an ice transiting speed of 7.21 knots. With a medium power installation of 100 mW intended for the 6 to 9-knot range, the use of 300 psf will give us 9.37 knots in the loaded condition and a slightly lower value, 9.09 knots, for 200 psf. In the empty condition, the corresponding speeds will be 9.71 and 9.74 knots respectively, with the speed increasing to 9.80 knots when only 100 psf is used. A high cushion pressure of 405 psf is only justified when a large amount of power (150 mW) is installed for higher ice transit speeds. We may now expect a speed of 13.0 knots both in the laden and empty conditions.

The above results apply to the continuous mode of operation in 7-ft thick level ice. Temporal and monetary constraints preclude a full exploration of the full possibilities inherent in the SSACV concept, but sufficient quantitative evidence has been produced to show that the use of the air cushion, even with a moderate amount of pressure, combined with the relatively thin side struts will be very effective in icebreaking and ice transiting, at least in the continuous mode of operation over 7-ft thick level ice.

TABLE V

Maximum Speed at Various Power Levels in Open Water and in 2.13 m thick Ice
(Speed in knots)

OPEN WATER	Calm Water Total installed power 70 mW	100 mW	150 mW	Sea State 5 Total installed power 70 mW	100 mW	150 mW
Laden	20.14	22.41	23.96	19.47	22.17	23.48
Empty - low strut immersion	22.78	24.72	27 ?	21.90	24.29	26 ?
- medium immersion	21.08	22.88	25.2 ?	20.37	22.52	24.30

ICE REGIONS (2.13 m thick ice)	Laden Condition Total installed power 70 mW	100 mW	150 mW	Empty Condition Total installed power 70 mW	100 mW	150 mW
Cushion pressure - 405 psf	3.21	9.13	13.0	3.21	9.13	13.0
	3.24	8.82	13.0	3.24	9.02	13.0
- 300	5.29	9.37	12.10	5.35	9.71	13.0
	4.92	8.80	12.0	5.26	9.23	13.0
- 200	6.15	9.09	11.24	6.80	9.74	12.5
	5.42	8.48	11.07	6.11	9.21	12.5
- 100	6.20	8.80	10.70	7.21	9.80	11.75
	5.52	8.20	10.48	6.55	9.46	11.91
No air cushion - 0	4.22	7.22	9.21	5.63	9.32	10.52
low strut immersion	4.19	6.82	8.97	5.54	8.36	10.64
medium immersion	-	-	-	4.83	7.81	9.80
				4.80	7.49	9.68

Note The upper and lower figures for the speed in the ice regions relate to the use of ice resistance equations 1 (Lewis and Edwards) and 2 (Edwards et al) respectively. The figures are very close.

It has also been hypothetically discussed in qualitative terms that the upper hull may be capable of cutting through elevated ridged-ice and, similarly, that the submerged hulls may be lowered clear of ridges below the uniform ice layer. Ridges constitute a major obstacle to the uniform progress of conventional icebreakers (or even ACV icebreakers) and a change of direction may be necessary in order to negotiate them. It appears that the SSACV type of icebreaker may have the unique capability of maintaining a straight course irrespective of the form of the ice layer (within some moderate limits) by changing its draught as necessary with the manipulation of the flexible seals. It is also similarly suggested that the icebreaking ability of the side struts may be improved by oscillating the vessel in pitch, again by the adjustment of the seals. However, this hypothetical behaviour can only be confirmed by model tests.

The SSACV Tanker could well be within the limitations of the state-of-the-art technology. The cushion pressure we are proposing is only of the same order as that used in the ACV-100 Icebreaker (1-1½ psi) and no special problems in containing this level of pressure are expected. However, there may be some problem areas and areas of uncertainty such as, for example, the Arctic applications of the technology now developed for some of the systems, particularly the behaviour of the flexible skirts under Arctic conditions and the problem of retracting the hemline of the skirt. The possibility of feeding air from the cushion to the outer surfaces of side struts is an attractive idea, but may require some engineering development. If successful, this supply of air will minimise strut-ice friction and also improve the steering qualities of the vessel in an ice environment by creating different hydrodynamic effects on the two sides of the struts. There is also the possibility

of using "puff ports", i.e. vents in the air cushion for producing a a side force on the vessel which will enable it to roll and clear sea room during ice transiting. This heeling effect may be combined with that due to the conventional method of shifting water ballast from one side of the vessel to the other. The small waterplane area of the side struts will only require the moving of a small quantity of water to produce a large angle of heel.

Thus, all in all, the combination of air cushion icebreaking with the mechanical side strut ice cutting appears to have commercial possibilities. If in addition we accept that the SSACV Icebreaker may be able to move unhindered through ice ridges, we will be adding another dimension to icebreaking and ice transiting activity in the Arctic regions. There is now no icebreaker, let alone an icebreaking tanker, operating in this type of environment and the time is now opportune for a fundamental change in concept in order to design a vessel with a significant improvement in icetransiting and general ridged-ice capability. The SSACV Icebreaking Tanker, in spite of its possibly higher cost and with some problem areas requiring further engineering development, could be the only solution to the problem of extracting and transporting LNG from Arctic regions on a year round basis and this concept could well be a candidate for the design of a Polar Class 10+ vessel.

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ADDENDUM to Notes on Design and Predicted Performance of the SNACV Icebreaking LNG Tanker (December 1981)

Page 3, Paragraph 2

Add at the end of the paragraph the following:

" This device will also reduce skirt wear and skirt friction over ice by providing air lubrication at the hemline. Also, the lowermost element may be considered as expendable and easily replaced when worn without affecting the main part of the skirt structure. "

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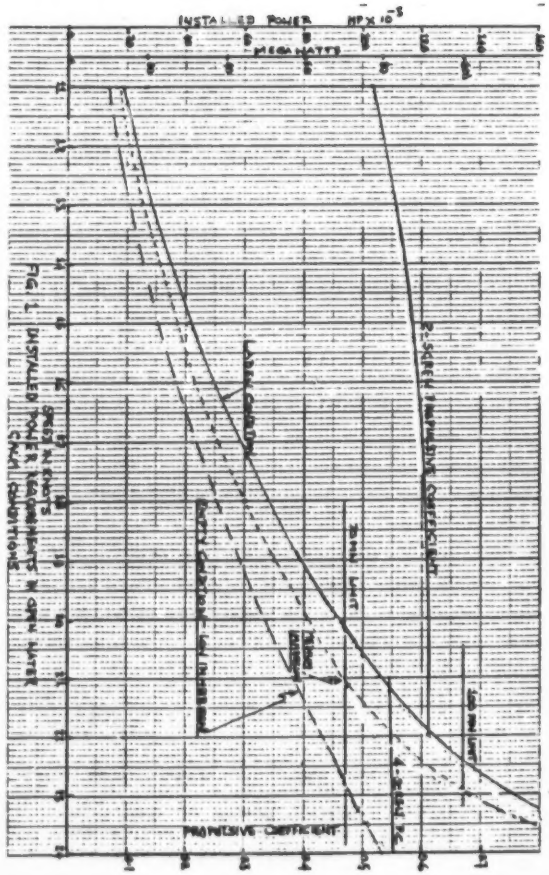

FIG. 1 INSTALLED POWER REQUIREMENTS IN OPEN WATER CRAFT CONDITIONS

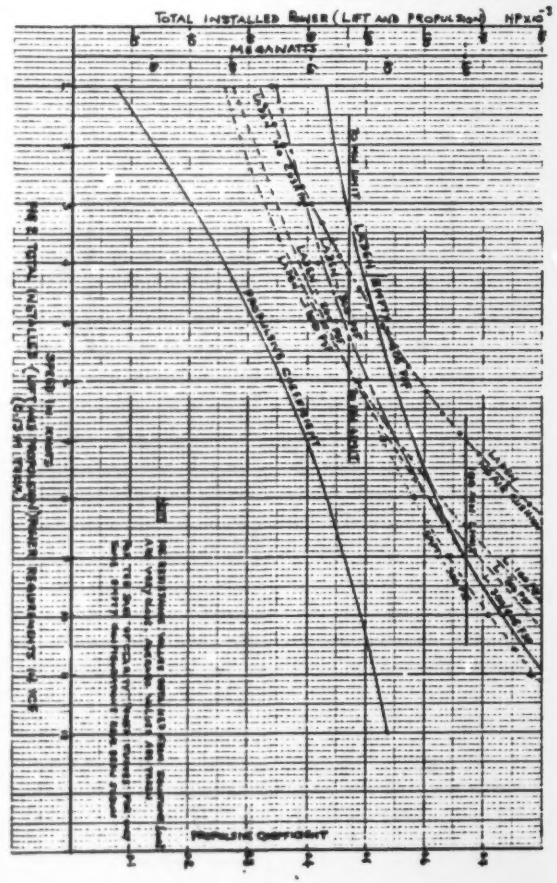

FIG. 2 TOTAL INSTALLED POWER REQUIREMENTS IN ICE

ANNEX A1

STONE VICKERS LIMITED

ANCHOR & HOPE LANE · LONDON SE7 7SZ

Our Ref: MAC/TB/C81H3477 14th December, 1981

Dr. T.K.S. Murthy
4 Birchington Close
Vinter's Park
Maidstone
ME14 5PF

Dear Dr. Murthy,

SSACV ICEBREAKING LNG/OIL TANKER

Using the data given in your letter of 5th December we have estimated the propeller performance particulars with the following results:-

1. Twin Screws for Open Water Speed of 22 kts

EHP = 72,000 HP.
BHP = 118,000 HP (total).
N = 90.
D = 9m.
V = 22 kts.
W_T = 0.286.
Bollard Thrust = 660 tonnes/shaft.
η_H = 1.15.
η_T = 1.0.

2. Quadruple Screws for Open Water Speed of 24 kts.

EHP = 97,000 HP.
BHP = 176,250 HP (total).
N = 165.
D = 9m.
V = 24 kts.
W_T = 0.17.
Bollard Thrust = 416 tonnes/shaft.
η_H = 1.02.
η_T = 1.3

I trust the above information is of use to you and its source will be acknowledged if you include it in your report.

Yours faithfully
for STONE VICKERS LIMITED

M.A. Clarke

M.A. Clarke
Technical Dept (Hydro)

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ANNEX A2

STONE VICKERS LIMITED

ANCHOR & HOPE LANE · LONDON SE7 7SZ

Our Ref: MAC/TB/C81H3477 21st December, 1981

Dr. T.K.S. Murthy
4 Birchington Close
Vinter's Park
Maidstone
ME14 5PF

Dear Dr. Murthy,

SSACV ICEBREAKING LNG/OIL TANKER

Further to my letter of even reference dated 14th December 1981, and our telecon of 17th December 1981, we have estimated the ice transit speeds of the vessel equipped with propellers designed for 22 kts and 24 kts open sea speeds:-

1. Twin Screws with 118,000 BHP installed
Ice transit speed = 4.8 kts.

$$\frac{EHP}{BHP} = 0.251$$

2. Quadruple Screws with 176,000 BHP installed
Ice transit speed = 6.8 kts.

$$\frac{EHP}{BHP} = 0.292$$

The propeller particulars used for the above are those given in my previous letter. I trust the information is of use in your investigation.

Yours Sincerely
for STONE VICKERS LIMITED

M.A. Clarke

M.A. Clarke
Technical Dept (Hydro)

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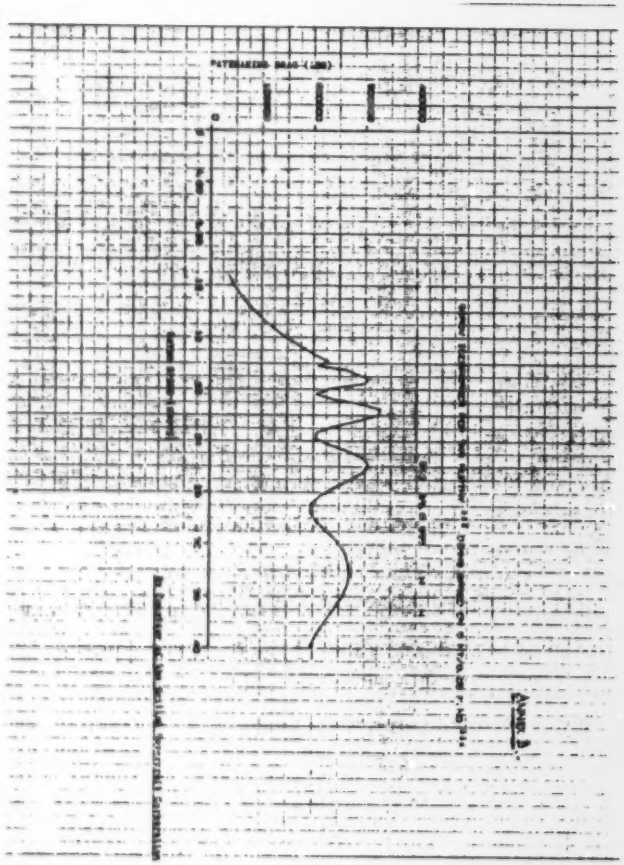

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